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Office of Research and Development
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Research Triangle Park, North Carolina 27711

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PROCEEDINGS OF THE SECOND STATIONARY SOURCE COMBUSTION SYMPOSIUM
Volume V. Addendum

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PROCEEDINGS OF THE SECOND STATIONARY SOURCE COMBUSTION SYMPOSIUM

Volume V. Addendum

Symposium Chairman Joshua S. Bowen
Vice-Chairman Robert E. Hall

Environmental Protection Agency
Office of Research and Development
Industrial Environmental Research Laboratory
Research Triangle Park, North Carolina 27711

Program Element No. EHE624

Prepared for

OFFICE OF RESEARCH AND DEVELOPMENT
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PREFACE

These proceedings document the more than 50 presentations and discussions of the Second Symposium on Stationary Source Combustion held August 29 -- September 1, 1977 at the Marriott Hotel in New Orleans, Louisiana. Sponsored by the Combustion Research Branch of the EPA's Industrial Environmental Research Laboratory-Research Triangle Park, the symposium presented the results of recent research in the areas of combustion processes, fuel properties, burner and furnace design, combustion modification, and emission control technology.

Dr. Joshua S. Bowen, Chief, Combustion Research Branch, was Symposium Chairman; Robert E. Hall, Combustion Research Branch, was Symposium Vice-Chairman and Project Officer. The Welcoming Address was delivered by Dr. John K. Burchard, Director of IERL-RTP; the Opening Address was delivered by Robert P. Hangebrauck, Director, Energy Assessment and Control Division, IERL-RTP; and Dr. Howard B. Mason, Program Manager NO_x Environmental Assessment Program, Acurex Corporation, delivered the Keynote Paper.

The symposium consisted of six sessions:

Session I:	**Small Industrial, Commercial and Residential Systems** Robert E. Hall, Session Chairman
Session II:	**Utility and Large Industrial Boilers** David G. Lachapelle, Session Chairman
Session III:	**Special Topics** David G. Lachapelle, Session Chairman
Session IV:	**Stationary Engine and Industrial Process Combustion Systems** John H. Wasser, Session Chairman
Session V:	**Advanced Processes** G. Blair Martin, Session Chairman
Session VI:	**Fundamental Combustion Research** W. Steven Lanier, Session Chairman

Volume V, Addendum, presents the Welcoming Address delivered by Dr. John K. Burchard, abstracts of the guest panelists, and those papers not recieved in time for the initial printing of the first three volumes. Appendix A gives a list of all attendees present at the Second Symposium on Stationary Source Combustion.

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The proceedings document the 50 presentations made during the Second Stationary Source Combustion Symposium held in New Orleans, LA, August 29-September 1, 1977. Sponsored by the Combustion Research Branch of EPA's Industrial Environmental Research Laboratory--RTP, the symposium dealt with subjects relating both to developing improved combustion technology for the reduction of air pollutant emissions from stationary sources, and to improving equipment efficiency. The symposium was divided into six parts, and the proceedings were issued in five volumes: Volume I--Small Industrial, Commercial, and Residential Systems; Volume II--Utility and Large Industrial Boilers; Volume III--Stationary Engine, Industrial Process Combustion Systems, and Advanced Processes; Volume IV--Fundamental Combustion Research; and Volume V--Addendum. The symposium was intended to provide contractor, industrial, and Government representatives with the latest information on EPA inhouse and contract combustion research projects related to pollution control, with emphasis on reducing nitrogen oxides while controlling other emissions and improving efficiency.

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WELCOMING REMARKS

By:

Dr. J. K. Burchard
Director, Industrial Environmental Research Laboratory - RTP
Environmental Protection Agency
Research Triangle Park, North Carolina 27711

Welcome to New Orleans. Some of you probably wonder --why New Orleans in August? That's a good question, one that I have asked too! Well, it's almost September and besides, you've got to admit, it's better than Chicago in January. Actually, we hope you will prefer to attend the meeting during the day and not wander outside until night when it's a little cooler.

As you know, this is the Second Symposium on Stationary Source Combustion. The first was held in Atlanta in September 1975. We hope to have the third one in the Spring of 1979.

Those of you who attended the last symposium will notice that the program has expanded. This is a 4 day meeting with about 50 speakers, compared to a 3 day meeting with about 35 speakers in Atlanta. This growth is due to the expanded interest in controlling pollution from combustion sources.

There are two major reasons for expanding the program. One is an awareness by EPA that to control combustion emissions for mobile sources would not be enough. Because of future growth and difficulties associated with reducing automobile emissions below certain levels, it will be necessary to tighten control of stationary combustion source emissions. The second reason relates to our country's need for energy independence. This means we will be burning coal in the future. Therefore, EPA must help determine how this can be done in an environmentally acceptable manner.

I won't dwell on the program since you will be provided with details over the next 4 days. However, I will touch on the highlights. The Combustion Research Branch is part of the Industrial Environmental Research Laboratory - RTP. For those of you who aren't familiar with our organization I'll describe it briefly.

I'll begin with IERL-RTP since Frank Princiotta will probably outline the structure of EPA's Office of Research and Development, of which we are a part, during his luncheon talk on Wednesday. He will also probably describe the Office of Energy, Minerals, and Industry, of which we are also a part.

Within IERL-RTP, there are three divisions and the Office of Program Operations. OPO, headed by Dr. John Smith, includes the Special Studies Staff and the Planning, Management and Administration Staff. One of the three divisions is the Utilities and Industrial Power Division, headed by Bill Plyler. Under it there are three branches: the Process Technology Branch, Emissions/Effluent Technology Branch, and

Particulate Technology Branch. The second Division is the Industrial Processes Division, headed by Chic Craig. It includes the Chemical Processes Branch, Metallurgical Processes Branch, and Process Measurements Branch. The third division is the Energy Assessment and Control Division, headed by Bob Hangebrauck, who will deliver the meeting's Opening Address. His division includes the Fuel Process Branch, Advanced Process Branch, and the Combustion Research Branch. It is the work of the last-mentioned branch that you are here to find out about in more detail.

The Combustion Research Branch is headed by Dr. Josh Bowen and consists of six engineers, four engineering technicians, and a secretary. Most of their work is contracted to private industry; however, there is also a small, but important in-house program. You will hear about both during the symposium.

I encourage you to read the pamphlet entitled "Controlling Pollution Through Combustion Research" which provides more detail about CRB's program. You should have received a copy with your registration package.

The main purpose of this meeting is to provide the public with results of recent combustion research sponsored by CRB, and to promote the exchange of ideas among the members of the combustion community. The agenda includes many of the nation's experts in the area of combustion research.

This morning's schedule includes the already mentioned Opening Address by Bob Hangebrauck of IERL and a Keynote Paper by Howard Mason of Aerotherm. This afternoon's session will cover small industrial, commercial, and residential combustion systems. The program for the remainder of the week includes: utility and large industrial boilers; a session on special topics, which didn't logically fall under any of the other sessions; a session on stationary engine and industrial process combustion systems; a session on advanced combustion processes; and a session on fundamental combustion research.

In order to provide comprehensive coverage, we have invited several guest speakers who have no direct connection with CRB's program. These include representatives from organizations such as the Electric Power Research Institute (to describe EPRI's NO_x control program), representatives of the Energy Research and Development Administration (part of the newly created Department of Energy) and representatives of private industry (who will discuss emerging combustion technologies), and representatives of EPA's Region V and the Office of Air Quality Planning and Standards (to discuss federal regulations). Dick Stern, Chief of the

Process Technology Branch of IERL, will review the status of flue gas treatment for NO_x and SO_x control, and a representative of TRW will describe the "Emissions Assessment of Conventional Combustion Systems" which is being sponsored by IERL's Chemical Processes Branch. A highlight of the meeting will be Wednesday's luncheon speaker, Mr. Frank Princiotta, who is Director of the Energy Processes Division of EPA's Office of Energy, Minerals and Industry.

The schedule has been arranged to include time for Questions and Answers dealing with the individual papers and for a General Discussion at the end of each session. Please feel free to ask questions or comment on our program.

Preprints of most symposium papers were provided with your registration packages. If time permits, you can look through papers of interest ahead of time. This will give you a chance to formulate questions before you hear the presentation.

I hope that you will enjoy the meeting and that this will be a worthwhile and productive 4 days.

SESSION I:

SMALL INDUSTRIAL, COMMERCIAL AND RESIDENTIAL SYSTEMS

ROBERT E. HALL
CHAIRMAN

EFFECTS OF FUEL AND ATOMIZATION ON NO_x CONTROL FOR HEAVY LIQUID FUEL-FIRED PACKAGE BOILERS

By:

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SECTION 1

SCOPE OF THE RESEARCH PROGRAM

Uncontrolled emissions from stationary fossil fuel-fired boilers are normally ranked coal-fired > oil-fired > natural gas-fired and this is reflected by the New Source Performance Standards for this class of equipment. However, if the same fuels were to be ranked either with regard to ease of control or to demonstrated successful application of a control technique, oil would rank below the other two fuels. Natural gas-fired combustors produce thermal NO and it is well-known that depression of flame temperature limits NO formation. Consequently, there are many successful examples showing the successful utilization of flue gas recirculation for the control of NO_x emissions from all types of natural gas-fired combustors. Coal contains significant quantities of bound nitrogen, and it is now recognized that under uncontrolled conditions the majority of the NO_x emission from pulverized coal-fired equipment can be attributed to fuel NO formation. NO_x reductions of the order of 65 percent can be claimed for combustion modification techniques with coal firing.

Residual fuels also contain bound nitrogen, normally less than coal, but combined flue gas recirculation and combustion modifications cannot claim consistent reductions greater than 70 percent. Staged combustion, the most effective method of controlling fuel NO production, requires the formation of a fuel-rich primary stage which encourages the formation of carbonaceous particulate which may not be completely burned out in the second stage with the consequent increase in particulate emissions. In many investigations involving staged combustion the maximum reduction in NO_x emissions is dictated by an increase in particulate emission. The formation of both of these pollutants are strongly linked to the details of the combustion process and the properties of the fuel. The program described in this paper has been planned to establish the effect of fuel properties and fuel atomization parameters on NO_x control techniques for package boilers fired with heavy liquid fuels.

The temperature-time-environment history of fuel and combustion products determines the fate of fuel-bound nitrogen, the rate of fixation of molecular nitrogen and the rate of destruction of nitric oxide by fuel fragments. In liquid fuel flames this history is controlled by the interaction of the liquid spray and the flow field which is composed of combustion air and recirculating combustion products. Liquid spray properties of importance are:

- dropsize distribution,
- droplet trajectory;
- droplet momentum, and
- spray pattern.

The characteristics of a wide range of commercial and specially designed atomizers will be tested under well-defined flow field conditions to assess the influence of the various parameters listed above on NO formation.

The nitrogen content of residual fuels is recognized as having a profound influence upon NO production. However, two other fuel oil properties will also have a strong effect on the final emission level, these are:

- fuel oil viscosity since this will control spray characteristics; and
- nitrogen evolution rate since this will establish when nitrogen compounds will enter the gas phase.

The total program can be divided into three distinct but interrelated investigations:

- Combustor experiments to assess atomizer and fuel effects.
- Spray characterization under cold and hot flow.
- Small-scale experiments to be conducted under controlled conditions to evaluate those parameters controlling fuel nitrogen formation in liquid fuel flames.

Spray characterization and the small-scale experiments will be carried out to explain and generalize the observations made in the combustor experiments.

The overall objective of the total program is to define atomization parameters which lead to satisfactory combustor operation, but minimize pollutant emissions. At the time of writing, results of these various studies were not available and the following discussion will be restricted to a description of the proposed effort and the rationale leading to the definition of the experimental program.

SECTION 2

COMBUSTOR EXPERIMENTS

Several studies have shown that fuel oil properties, particularly nitrogen content, and atomizer design influence the emission of pollutants from oil-fired combustors. The objective of the combustor experiments is to systematically compare atomizer performance for selected air flow conditions with a broad range of fuels. The fuel screening and atomizer performance investigations will be carried out in two combustors whose characteristic dimensions are typical of watertube and firetube package boilers. One important factor in the design of burners for oil firing is the compatibility of the air pattern and the liquid fuel spray distribution. The combustor experiments will be carried out with a burner capable of providing precisely repeatable conditions at the throat. All the atomizers selected for screening will be tested with at least two air distributions which are known to have different effects upon the fuel/air mixing pattern.

ATOMIZER DESIGN AND POLLUTANT FORMATION

The basic principle of disintegration of a liquid into a spray requires that there be a continual change in the form of the liquid such that the total surface area is increased until the liquid volume becomes unstable and disintegration occurs. This instability is generated by an initial forcing function causing extreme acceleration of portions of the liquid. The droplet size distribution of the resulting stream depends upon the nature of flow in the atomizer, the physical properties of the liquid, the state of the ambient atmosphere, and the source and manner in which energy is imparted to the liquid to provide the forcing function. It is convenient to classify atomizers according to the source of energy used for the atomization process and the majority of atomizers fall into liquid prssure, centrifugal or assist-fluid energy categories. Several example of liquid pressure and assist-fluid atomizers will be tested as part of this program, together with atomizers which fall outside these categories.

Atomizers may also be classified by the general type of spray pattern they produce.

- Full core sprays; the whole conical volume of the spray is filled with droplets.
- Hollow core spray; the droplets are concentrated on the outer periphery of the conical volume and the core is relatively free of droplets.
- Flat or fan sprays; the droplets are distributed along a narrow wedge-shaped band.

The hollow core spray is used for most combustion applications; however, several full core sprays will be used in this investigation to assess the effects of spray distribution.

Cato et al (1) reported the influence of various atomizer designs on boiler efficiency, particulate emissions and NO_x emissions from a series of field operating boilers. No systematic influence of the type of the assist-fluid could be ascertained, probably because any effect of the atomizer was masked by the variations in other parameters. In one series of investigations with staged combustion the use of steam as the assist-fluid produced lower emission levels than those obtained with air. The effect of atomizer properties under staged combustion conditions are illustrated by the results presented in Figure 1. The same assist-fluid atomizer was used with a staged burner, the second stage air was injected from the periphery of a short primary combustion chamber. The fuel oil contained 0.71 percent nitrogen and there is strong evidence that with this particular system better results were obtained with steam as the atomizing fluid.

Oil spray properties can be varied by using the same nozzle design but different capacities and operating at a constant oil throughput (and therefore different delivery pressures). Figure 2 presents NO_x emissions as a function of swirl level of the combustion air for three different nozzle capacities. It can be seen that at low swirl levels there is a very significant effect of fuel pressure which might be associated with either droplet size distribution or droplet penetration or a combination of these effects.

The results presented in Figure 3 show the influence of both atomizer design and fuel oil throughput on particulate emissions from a refinery furnace. Particulate emissions are greatest with pressure jet atomization with a high fuel pressure and lowest with an L-type steam atomizer. Characteristic droplet size distributions for the same nozzles are presented in Figure 4 showing that there is not a simple relationship between droplet size distribution and particulate emission. Fuel/air mixing will also influence particulate formation. It can be suggested that droplet distributions which have steep maxima will tend to concentrate the fuel at a given point. Efficient fuel/air mixing can be achieved by having a relatively flat droplet distribution curve and relying upon their different trajectories to distribute the fuel in the air stream.

Both of the examples of the effect of nozzle parameters on pollutant formation presented in Figures 2 and 3 illustrate the strong coupling of fuel spray-flow field interaction in liquid fuel oil flames.

FUEL OIL PROPERTIES AND POLLUTANT FORMATION

Fuel oil properties will influence pollutant emissions for three major reasons; these are:

1. Bound nitrogen content – It is well-known that NO emissions increase with increasing fuel nitrogen content; however, the fraction of the bound nitrogen converted to NO decreases with increasing nitrogen content.
2. Asphaltene content – Certain workers (4) have reported a linear relationship between fuel oil asphaltene content and particulate emissions. This might well be an oversimplification resulting from the use of fuel from the same crude source and it is unlikely to be a universal relationship. Munroe and Watson (5) report a poor correlation of particulate emissions with asphaltene content.
3. Oil viscosity – Fuel oil viscosity will affect the fuel spray which will affect the fuel/air mixing process which could be reflected in a change in pollutant formation.

Figure 5 which is similar to Figure 2 shows the effect of fuel oil temperature on NO emissions as a function of swirl level. The results presented in Figure 6 were obtained by Jackson and Cunningham (6) in a fuel oil-fired test rig show that as the fuel temperature is decreased and the viscosity increases, particulate emissions increase and NO and SO_3 emissions decrease. Changes in fuel oil temperature will vary the spray characteristics because as the temperature changes both the viscosity and delivery pressure must change for a constant oil flow. Droplet temperature may also affect fuel vaporization rates which will affect the overall heat release profile.

There have been many examples of the effect of total fuel oil nitrogen content on NO formation in liquid fuel-fired combustors. Although the estimated fuel nitrogen conversions have similar characteristics for the various investigations there is considerable scatter about any mean line. This scatter may be associated with the details of the combustion system or it may be attributable to properties other than total fuel nitrogen content which control fuel NO production. These properties may be associated with the source of the crude, details of the refinery process or the consequence of blending with lighter fuels to produce a low sulfur fuel. During the conduct of this program we intent to test, under controlled conditions, a series of residual fuels representative of crudes used throughout the country. Each fuel will be tested with several atomizers and at various delivery temperatures.

SECTION 3

SPRAY CHARACTERIZATION

The interpretation of the results obtained in the combustor investigation will require some characterization of the sprays produced by the various atomizers and the effect of the flow field on the spray distribution. The atomizers used in the combustor experiments will be placed in a cold flow test rig and supplied with liquid of the same viscosity as the fuel oil. The cold flow tests will be carried out in a chamber with the nozzles fitted in a burner throat identical to the ones used in the combustor investigations. Therefore, the properties of the spray will be determined under conditions similar to those encountered under combustion conditions. The flow field will be modified somewhat because of the different density field.

Two methods will be used to assess spray characteristics:

- Droplet size distribution will be measured by a laser diagnostic technique developed by Swithenbank et al (7). The technique is based on the Fraunhofer diffraction of a parallel beam of monochromatic light by the moving droplets. A Fourier transform lens is used to focus a stationary light pattern onto a multi-element photo detector to measure the diffracted light energy distribution. This light energy distribution is translated into the corresponding unique droplet size distribution and the droplets are classified into 31 size groups spanning two decades of diameter. The accuracy of the method is estimated to be approximately 2 percent of the characteristic diameter.
- Physical sampling will, under isokinetic conditions, be used to provide the general liquid mass distribution as a function of distance from the nozzle.

A limited number of measurements will be attempted under combusting conditions to approximately evaluate the spray path to compare with the cold flow measurements. The spray characterization tasks will be carried out at the IFRF Research Station in Holland.

SECTION 4

SMALL-SCALE EXPERIMENTS

NO emissions measured in the combustor investigations will be the sum of both fuel and thermal NO and consequently, the results will be difficult to interpret. A series of small-scale experiments are planned to establish those parameters which control fuel nitrogen conversion in fuel oil flames. Three techniques can be employed in an attempt to establish the relative contributions of fuel and thermal NO formation to the total emission, these are:

- Addition of cooled combustion products to the combustion air supply. This will reduce flame temperatures and minimize thermal NO production. However, it will not positively eliminate thermal NO production and the increased mass of oxidant plus diluent will probably influence the fuel/air mixing process which could change the fraction of fuel nitrogen converted to NO.
- Subtraction of NO produced by a non-nitrogen-containing fuel. This method has the advantage of being relatively easy to set up, but the properties of the non-nitrogen liquid fuel will very different from the residual fuel, and it is most unlikely that the thermal NO production will be identical for the two liquid fuels.
- Substitution of molecular nitrogen in the air by argon to eliminate thermal NO production. With this method it is certain that all the NO emitted results from the oxidation of fuel-bound nitrogen. The major uncertainties in the assessment of fuel NO are associated with temperature effects and any chemical effects due to the almost complete lack of molecular nitrogen. The former difficulty can be overcome by diluting the argon with CO_2 to provide the same theoretical flame temperature as would be obtained with air.

The latter approach has been chosen since it is believed that the use of argon/CO_2 diluent mixtures will give the most accurate assessment of the parameters controlling fuel NO formation. The tests will be carried out at subscale because of the prohibitive cost of argon substitution in either of the package boiler simulators.

The overall purpose of the subscale testing is to provide information which will improve our basic understanding of the processes associated with the formation of fuel and thermal NO_x in oil-fired systems. This information will enable the results from investigations with the package boiler simulators to be generalized. To achieve this basic goal we believe that two types of experimental studies will be required (see Table I).

- Phase I – NO_x Formation in Swirling, Turbulent Diffusion Flames

  The overall goal of this first phase is to define the importance of fuel nitrogen conversion to NO in residual oil flames. Use of typical combustion hardware and direct simulation of actual practice will be emphasized. (This is in contrast to Phase II in which the emphasis is placed upon creating specific, ideal types of combustion environments). In general, the approach in Phase I is to conduct a relatively large number of experimental tests with air as the oxidizer and then, based on detailed consideration of the results, select specific conditions for testing with $Ar/O_2/CO_2$. The work is divided into two Tasks:

  - Spray Characteristics, and
  - Fuel Chemistry.

  The purpose of Task 1.1, Spray Characteristics, is to define the influence of drop size and spray pattern on the formation of fuel and thermal NO. Since these parameters are generally coupled, several types of tests will be conducted. These results, together with the spray characterization data, should allow the influence of the controlling parameters to be ascertained. Current plans are to include four types of mechanical atomizers (hollow and solid cone, wide and narrow angle) an air-assisted sonic nozzle (the same nozzle type will be used in three sizes of combustor) and a special design of twin fluid nozzle. The viscosity at the nozzle will be altered by varying the oil preheat temperature. Atomization pressure will be studied in conjunction with nozzle size at a constant fuel input rate. Most of the spray characteristics studies will be carried out with two fuels of widely different characteristics.

The purpose of Task 1.2, Fuel Chemistry, is to establish the influence of the oil composition on the formation of fuel and thermal NO.

At least three composition parameters (in addition to total nitrogen content) are of potential importance:

- Hydrocarbon distribution as it influences the physical properties of the oil, e.g., viscosity.
- Relative location of the nitrogen and hydrocarbon components on the distillation curve.
- Amount and type of fuel sulfur present.

Several residual oils, a distillate oil, kerosine, pure hydrocarbons, and various sulfur and nitrogen compounds will be used in this phase of the investigation. The effect of the addition of both a light and a heavy aliphatic primary amine to a pure compound fuel will be studied to define the importance of relative volatilaties. Mixed nitrogen additives will also be considered to establish whether the presence of early formed NO inhibits the oxidation of refractory nitrogen specie.

Current plans are to use a single atomizer for most of the chemistry studies. Limited testing under alternate atomization conditions will be conducted to demonstrate the generality of the conclusions. As part of this testing we will also examine the effects of overall combustion parameters (wall temperature/heat loss, local gas temperature, and combustion aerodynamics) but detailed consideration of these parameters will be left until Phase II.

- **Phase II — Controlled Mixing History Studies**

  The overall goal of the second phase is to define those conditions which produce the minimum NO_x emissions. In this phase the focus is less on simulating current practice and more on conducting completely controlled experiments. Detailed studies would be conducted on the influence of:

  - initial droplet heating rate,
  - rate of oxidant addition,
  - use of secondary fuel injection, and
  - temperature history.

These small-scale studies will be carried out in a refractory down-fired furnace capable of firing at 100,000 Btu/hr. The furnace is constructed from three modular sections with a central 8-inch diameter test chamber. Wall temperature control is provided by four auxiliary heating chambers fired by four propane burners. A schematic of the furnace system is presented in Figure 7. The fuel and oxidant delivery system will allow preheated air or argon CO_2 mixtures to be delivered to the burner which can fire oil diluted with either or both of two dopants to vary the nitrogen and sulfur content of the fuel.

SECTION 5

CONCLUSION

A program has been planned which will allow the systematic investigation of the effect of atomizer design and fuel oil properties on the formation of NO_x in package boiler combustion chambers. The total program is divided into three sections:

Combustor Investigations — Tests will be carried out with a cross section of available fuel oil atomizers and well-defined burner flow fields to establish the influence atomizer design on pollutant formation. The investigations will be carried out with various fuels in combustors whose configurations are typical of package boilers.

Spray Characterization — Atomizers used in the combustor investigations will be characterized under cold and hot flow conditions to assess dropsize distribution on the effect of air flow on droplet penetration.

Small-Scale Experiments — A special purpose furnace is being constructed to investigate the factors affecting fuel NO formation.

The overall goal of this program is to specify atomizer design parameters for use in low NO_x combustors using distributed air addition or staged heat release with remote air injection.

REFERENCES

1. Cato, G.A., Muzio, L.J. and Shore, P.E., "Field Testing: Application of Combustion Modification to Control Pollutant Emissions from Industrial Boilers, Phase II", EPA Report No. 600/1-76-086a, April 1976.

2. Cichanowicz, J.E. and Heap, M.P., "Design Criteria for Low NO_x, High Efficiency Oil-Fired Firetube Package Boilers", Report in preparation, EPA Contract 68-02-1500.

3. Van Breugal, J.W., Hasenack, H.J.A., Spink, C.D., "Soot and Solid Emissions from Oil-Fired Heater with Swirling Air Burners", paper presented at the 4th Joint Meeting of the Gas and Oil Panel, September 1976, IJmuiden, Holland.

4. Cheetham, H.A. and Champion, J.B., "Development of Large Burners for Power Station Boilers, Paper 110, Deuxiéme Symposium Europeen sur la Combustion, Orleans, France, 1st-5th Sept., 1975.

5. Munro, A.J.E. and Watson, C.D., "The Interaction of Fuel Oil Quality and Equipment Performance on Emissions from Industrial Oil-Fired Steam/Hot Water Boilers", paper presented at the 4th Joint Meeting of the Gas and Oil Panel, September 1976, IJmuiden, Holland.

6. Jackson, P.J. and Cunningham, A.T.S., "The Combustion of Residual Fuel Oil", paper presented at the 4th Joint Meeting of the Gas and Oil Panel, September 1976, IJmuiden, Holland.

7. Swithenbank, J., Beér, J.M., Taylor, D.S., Abbot, D. and McCreath, G.C., "A Laser Diagnostic Technique for the Measurement of Droplet and Particle Size Distribution", paper presented at the AIAA 14th Aerospace Sciences Meeting, Washington, D.C., January 1976.

TABLE I. OUTLINE OF RESEARCH PLAN FOR SMALL-SCALE STUDIES

Phase I — NO_x Formation in Swirling, Turbulent Diffusion Flames

1.1 Spray Characteristics

- viscosity/oil temperature
- atomization pressure
- atomizer type

1.2 Fuel Chemistry

- 10 residual oils
- distillate oil, vaporizing oil
- pure compound fuels
- nitrogen doping
- sulfur doping

Phase II — Controlled Mixing History Sutdies

- oxidant addition pattern
- fuel addition pattern
- initial heating rate
- time/temperature history

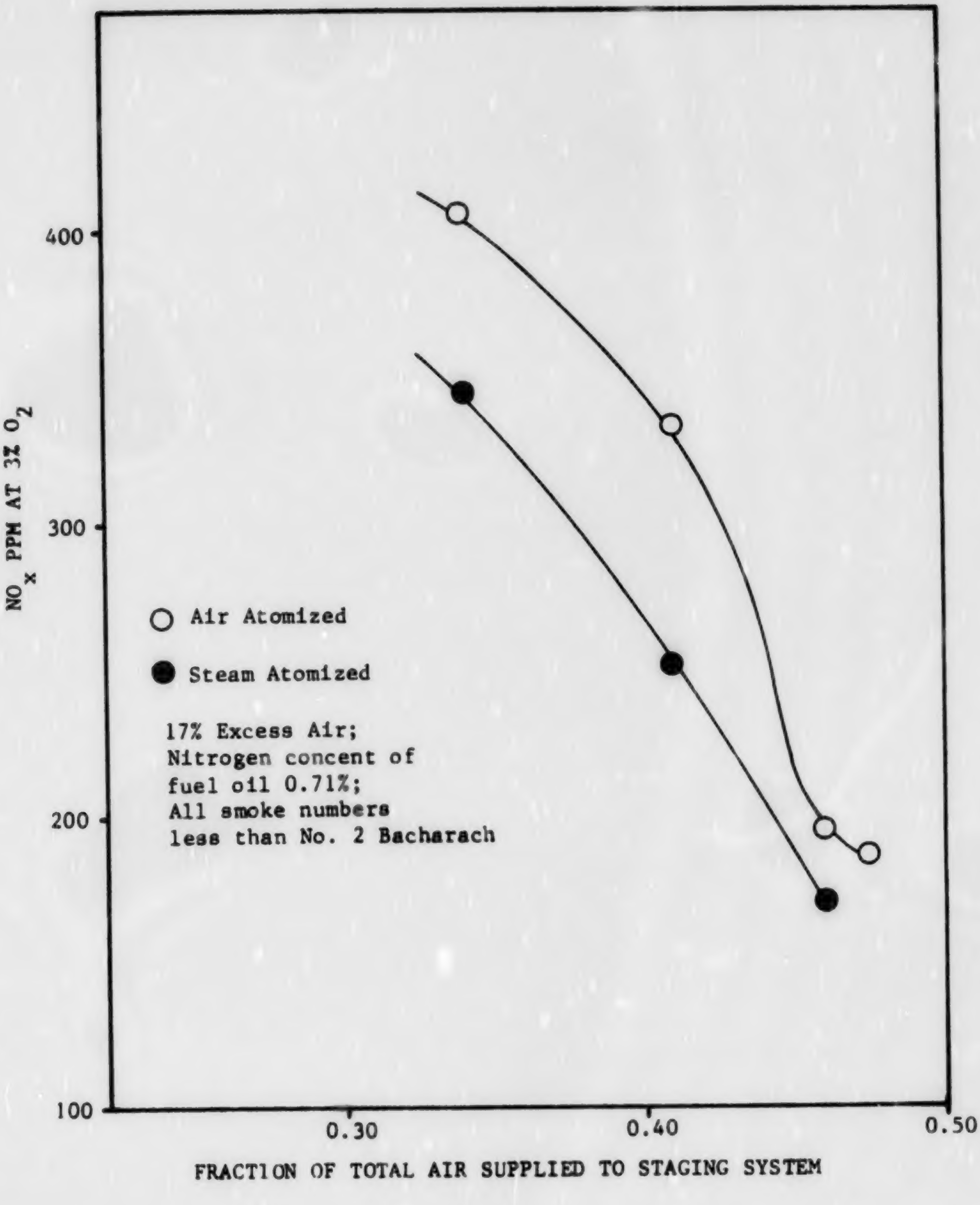

Figure 1. The effect of atomization fluid on staged combustor performance (2)

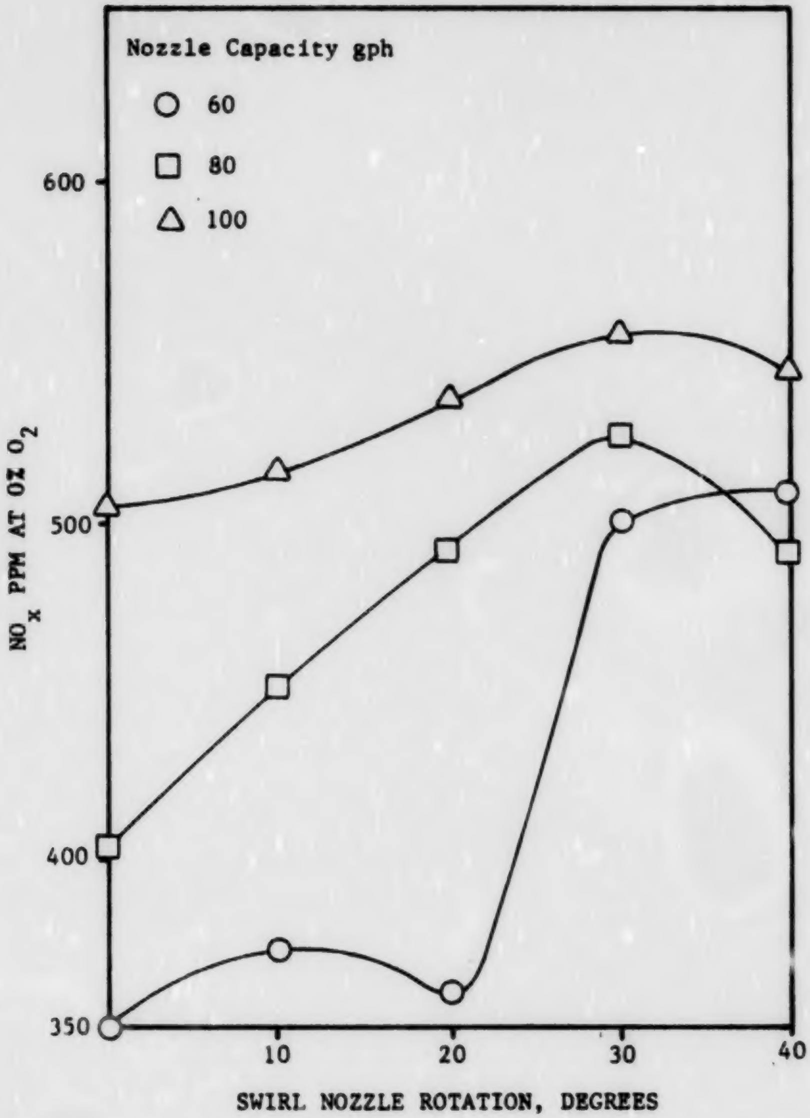

Figure 2. The influence of nozzle size on NO formation at constant oil throughout (2) (steam assist atomizer, 3 x 10^6 Btu/hr input, 17% excess air).

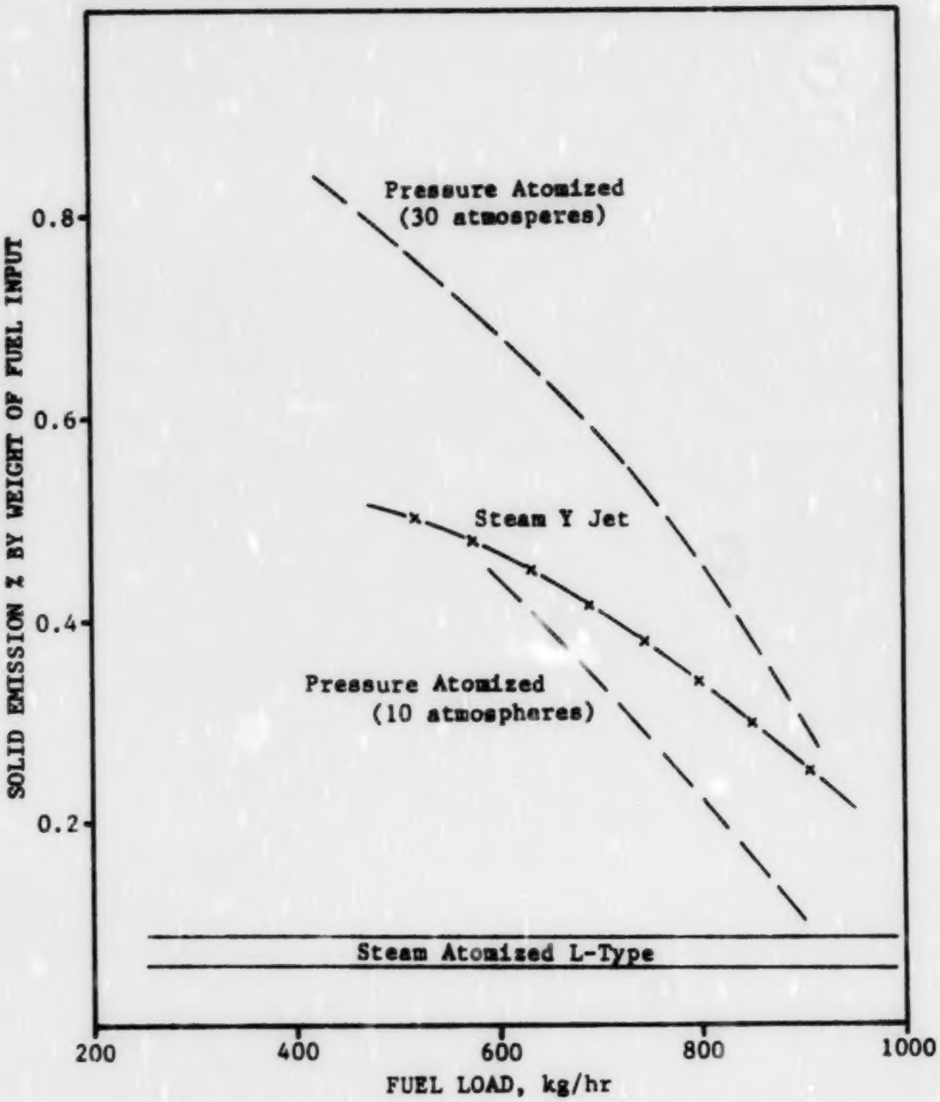

Figure 3. Solid emissions from a refinery heater as a function of fuel load (3)

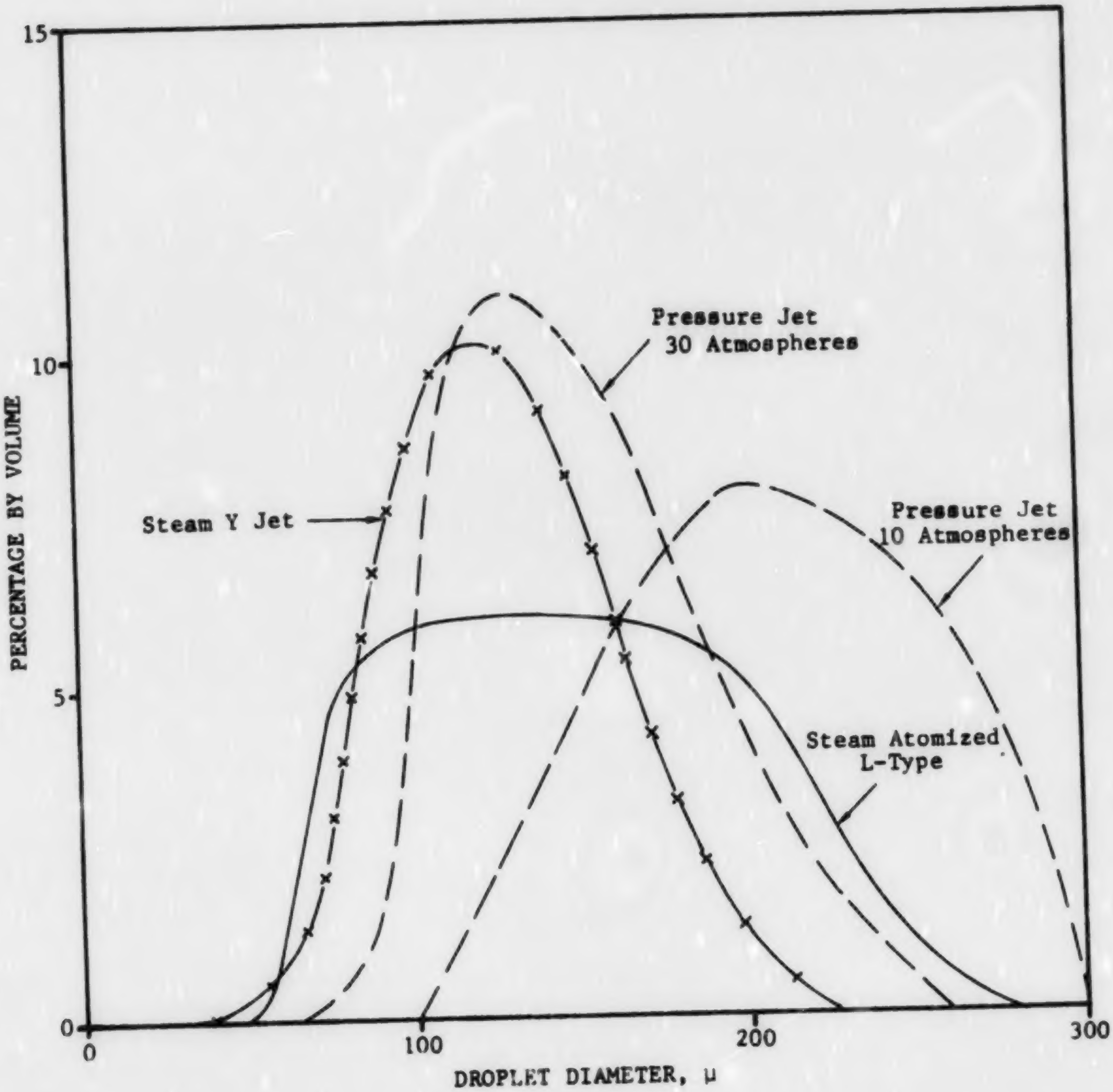

Figure 4. Dropsize distribution for the atomizers used in Figure 3 (3).

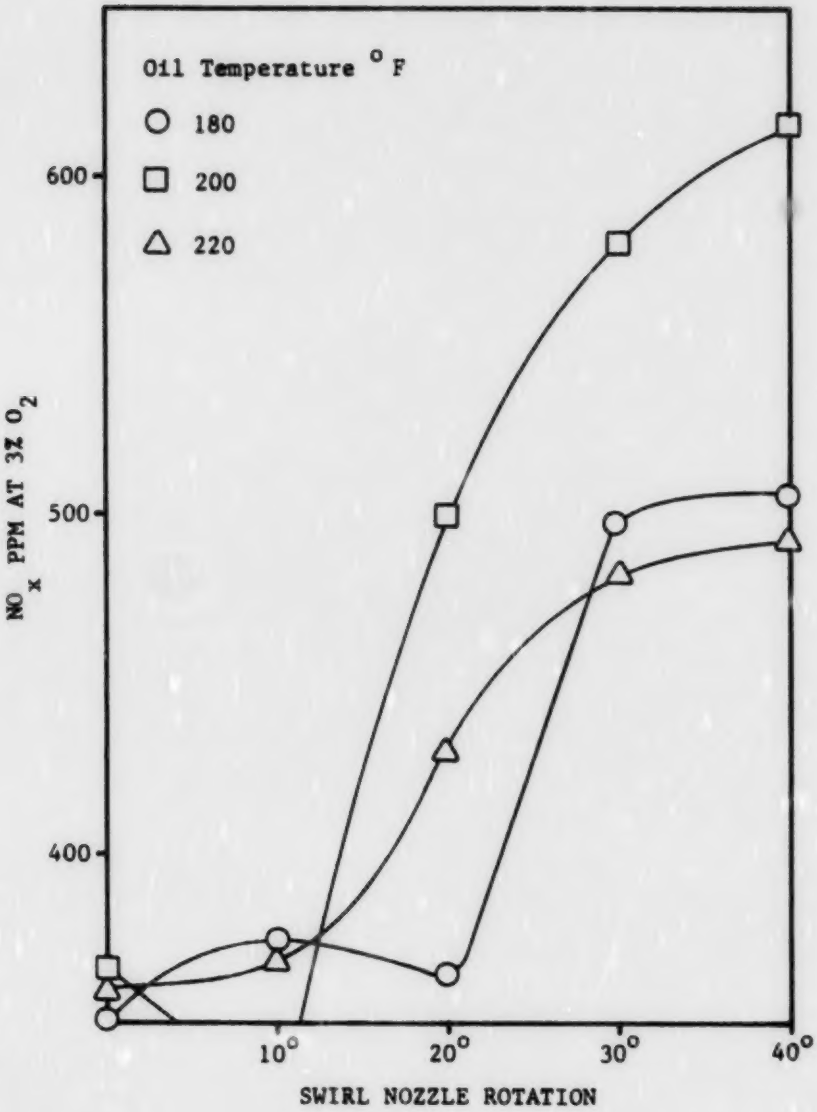

Figure 5. The effect of fuel oil temperature on NO_x emissions (2).

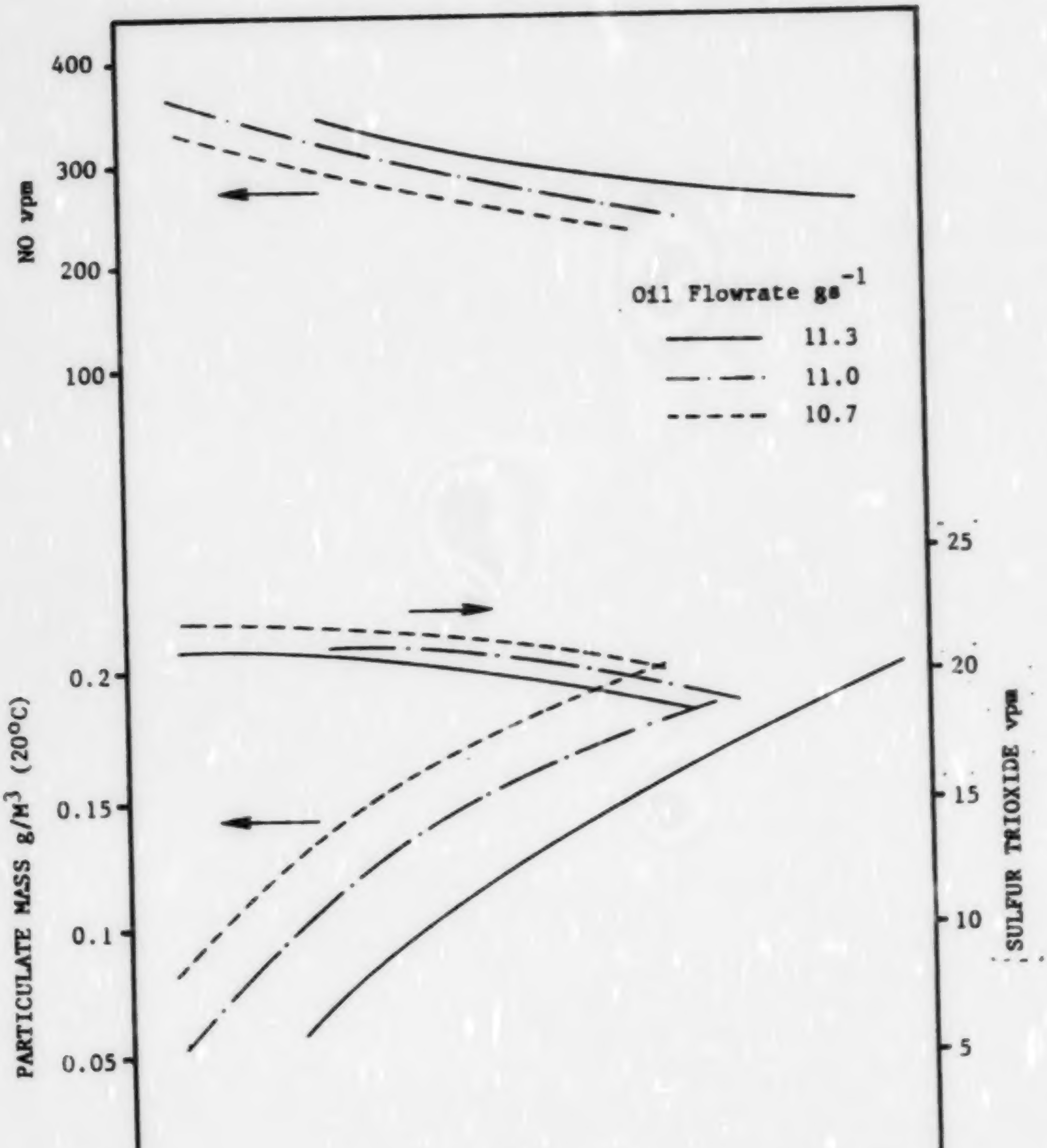

Figure 6. The effect of firing rate and oil viscosity on pollutant formation (6).

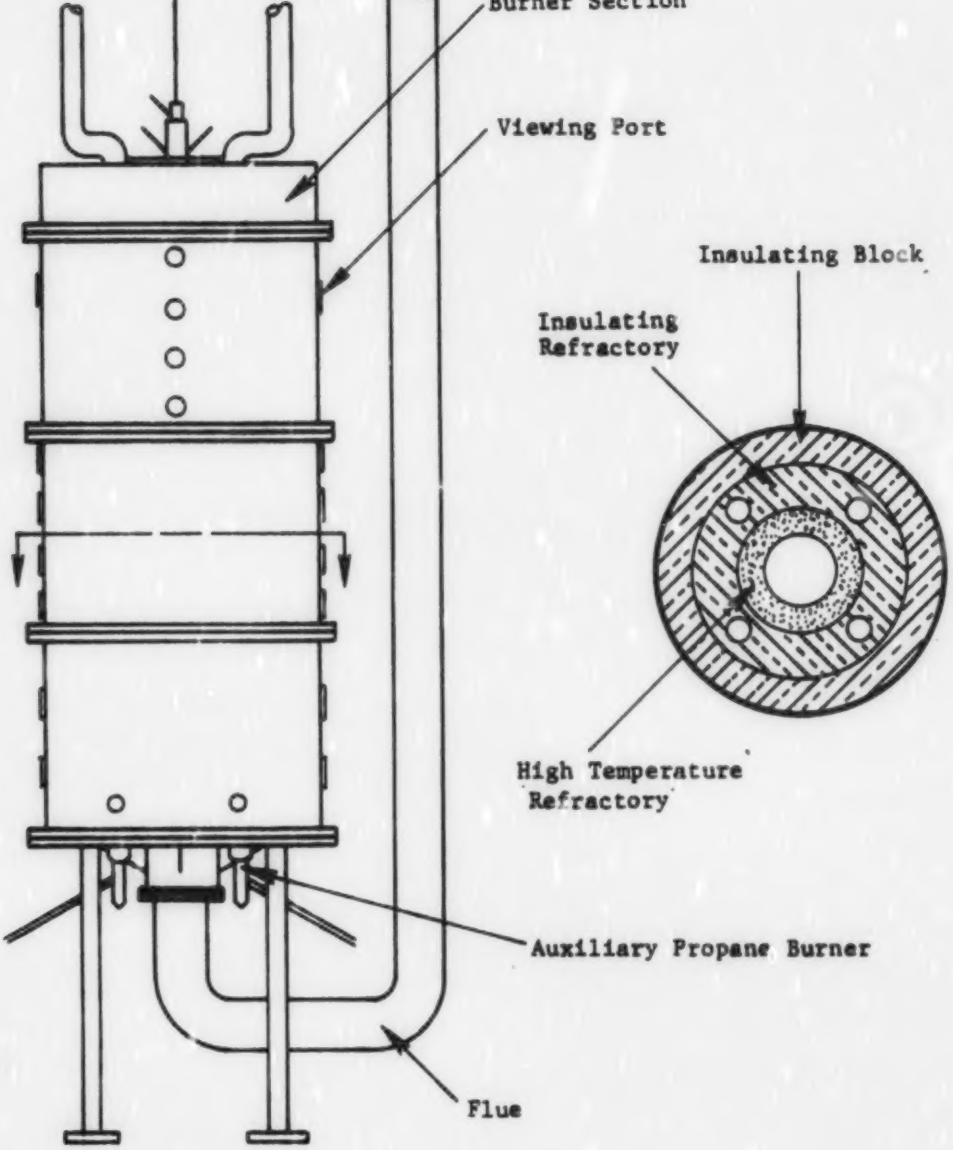
Burner Section
Viewing Port
Insulating Block
Insulating Refractory
High Temperature Refractory
Auxiliary Propane Burner
Flue

Figure 7. Argon Furnace,

NO_x CONTROL TECHNIQUES FOR PACKAGE BOILERS: COMPARISON OF BURNER DESIGN, FUEL MODIFICATION AND COMBUSTION MODIFICATION

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SECTION 1

INTRODUCTION

An arbitrary but useful method of categorizing boilers is to assign a nominal firing rate range to particular uses. Industrial boilers are normally considered to be those boilers with a design firing rate between 10 x 10^6 and 500 x 10^6 Btu/hr. In the upper end of the size range they are virtually indistinguishable from utility boilers. At the opposite end of the size range their characteristics are the same as commercial boilers. Commercial boilers can be considered to be represented by those boilers with a design firing rate of between 0.33 x 10^6 and 10 x 10^6 Btu/hr. Boilers represent significant sources of three atmospheric pollutants; nitrogen oxides (NO_x), sulfur oxides (SO_x) and particulates. This paper is primarily concerned with a review of those techniques available to control the emission of one pollutant, NO_x, from industrial and commercial boilers.

It has been estimated (1) that boilers account for greater than 60 percent of the NO_x emitted annually by stationary sources in the U.S. Recently Barrett and Putnam (2) have estimated the number of boilers in use in the Continental United States as a function of design firing rate. Making allowance for the fact that no single data source was complete, National Emissions Data Bank (NEDS) and American Boiler Manufacturers Association data were used together with EPA emission factors to estimate the annual cumulative NO_x emissions for all boilers based upon their actual use. Figure 1 shows the fraction of the total NO_x emissions attributable to boilers as a function of boiler design firing rate and fuel capability. Commercial and industrial boilers apparently account for about 50 percent of the emissions and oil firing is the most significant producer of NO_x for these boilers, although the importance of coal increases as the boiler size increases. The data presented in Figure 1, which is based upon information presented by Barrett and Putnam (2), suggests that the contribution of industrial boilers to the total emission of nitrogen oxides is more significant than had been indicated by other studies. Mason and Shimizu (3) indicate that emissions from commercial and residential, industrial and utility boilers are in the ratio of 1:2.4:6.6, whereas Figure 1 indicates a ratio of 1:3:3. These differences most probably reflect uncertainties in the estimates of load factors used to derive the total emissions. Seemingly, total NO_x emissions from industrial boilers are comparable to those from utility boilers, yet the potential to control NO_x emissions from industrial boilers has not been investigated as exhaustively as it has for utility boilers. This can be readily explained by the much smaller number of sources which account for the utility emissions making control more feasible. However, as emissions from utility boilers are reduced by the application of control techniques, uncontrolled emissions from industrial boilers will become more significant and economical means should be sought to control NO_x emissions from this class of equipment.

The industrial boiler population comprises several different design types and includes the utilization of the complete range of fossil fuels. They are normally classified by their design type (watertube or firetube) and their size, which is normally described in boiler horsepower, steam generated per hour or hourly heat output (one boiler horsepower is approximately equal to 33.5 lbs of steam per hour or 33,500 Btu/hr heat output). In the small boiler sizes (25M Btu/hr) gas is the dominant fuel and firetube boilers are the dominant design type. As the name implies, the major characteristic of this design is that the products of combustion pass through tubes surrounded either by water or steam. As the firing rate increases the dominance of gas decreases and the boilers are predominantly of the watertube type. In this type of construction water flows through tubes thus reducing its total volume, and the design is normally favored over firetube boilers for pressures in excess of 150 psig. A detailed description of the design characteristics of various commercial and industrial boilers is given by Offen et al (4) and a breakdown of the industrial boiler population by Locklin et al (1).

It is well-known that nitrogen oxides can be produced from two sources during the combustion of fossil fuels, molecular nitrogen and nitrogen chemically bound in the fuel. The selection of an emission control technique for a particular boiler type will primarily depend upon the source of the nitrogen accounting for the major portion of the total emission of nitrogen oxides. Techniques which have been successfully demonstrated for the control of thermal NO production and which depend solely on flame temperature reduction will have a minor influence upon fuel NO production. Also, a control technique which has been optimized for fuel NO control for a given fuel will probably not be the optimum system if the fuel were to be changed.

Three methods of controlling pollutant emissions from package boilers are compared in the paper with respect to both thermal and fuel NO control. These three methods are:

- Fuel selection
- Burner optimization
- Combustion modification.

The results included in the following sections were obtained under three separate contracts concerned with the development of NO_x control techniques for package boilers (5, 6, 7). The reader is directed to other publications for a complete description of the relevant hardware which will not be described in detail below.

SECTION 2

FUEL SELECTION TO CONTROL NO_x

Fuel properties will influence the emission of NO_x from a given system for two major reasons:

- flame temperature. Thermal NO production is strongly linked to the maximum attainable flame temperature which will be dependent upon the fuel composition; and
- nitrogen content. Conversion of fuel-bound nitrogen will account for some fraction of the total NO_x emission from a given system; thus emissions will increase as the fuel nitrogen content increases.

Alcohol fuels do not contain bound nitrogen and they have lower flame temperatures than natural gas or residual fuels. Consequently, conversion to this type of fuel would have a beneficial effect upon emissions for the two reasons listed above.

Martin (8) compared the emissions of NO_x from a laboratory furnace burning methanol and a number of other conventional fuels and the results indicated that methanol emissions were lower, consistent with the lower well-mixed adiabatic flame temperature. The effect of fuel type on NO_x emission from two field operating industrial boilers is shown in Figure 2. These results were obtained in a recent series of field tests (6) from a 12K lbm steam/hr firetube boiler and a 25K lbm steam/hr watertube boiler and show the NO_x emissions (corrected to 0% O_2) as a function of excess oxygen level. The No. 5 fuel oil contains 0.1 percent N, thus the higher emission levels and greater sensitivity to excess air level are at least partially due to the presence of fuel nitrogen compounds. Emissions from methanol are significantly lower than those from natural gas, although the difference is smaller in the case of the watertube boiler. This can probably be attributed to the lower flame temperatures in the watertube boiler due to the higher heat extraction rates and lower combustion intensities. Due to limitations of the methanol delivery flowrate the comparative tests were carried out at a steam load that was 30 percent of the design capacity of the watertube boiler and 80 percent capacity of the firetube boiler.

SECTION 3

COMBUSTION MODIFICATION FOR NO_x CONTROL

The combustion process can be modified in two ways to control the formation of NO:

- flame temperature can be reduced by dilution of the reactants with inerts either by adding cooled combustion products to the combustion air supply or by the entrainment of recirculating combustion products within the combustion chamber. Alternatively, peak flame temperatures can be reduced by staged heat release with interstage cooling;
- conversion of fuel-bound nitrogen is most strongly dependent upon oxygen availability. Consequently, by staging the heat release to give a fuel-rich primary zone the formation of fuel NO is limited. In practice this is achieved by dividing the total air supply into two and injecting one of these streams downstream from the point of fuel injection.

Both of these control techniques have been investigated extensively in utility and large industrial boilers. This section discusses their application to package boilers.

The value of flue gas recirculation (FGR) as an NO_x control technique is particularly dependent upon the nitrogen content and the fuel. Figure 3 presents results comparing the NO_x emissions from two field operating package boilers as a function of the mass of recirculated flue gas. Significant reductions were obtained with the nitrogen-free fuels while the reductions with nitrogen-containing No. 5 fuel oil were marginal. The marginal reductions obtained with No. 5 fuel oil are consistent with the results of other investigators who have demonstrated the relative ineffectiveness of FGR as a control technique for fuels containing bound nitrogen. Figure 4 provides a comparison of the results achieved by several workers in different systems.

The addition of cooled combustion products to the combustion air supply will increase burner momentum which will affect fuel/air mixing. Variations in fuel/air mixing might well enhance fuel NO formation, thus giving a net increase in emissions even though a reduction in thermal NO production were to be achieved. Increases in NO_x emission with flue gas recirculation were observed under some operating conditions with the watertube boiler during the field tests described by Cichanowicz et al (6) and in a series of tests with a laboratory combustor (7). The range of variation in the results from both of these tests is shown in Figure 4. Figure 5 shows the effect of FGR for non-nitrogen-containing fuels. Comparison with Figure 4 illustrates how much more effective FGR is as a control technique for fuels which do not contain bound nitrogen.

Figure 6 compares the results of several investigations showing the reduction in NO_x as a function of burner stoichiometry for a variety of combustion systems and staging methods for nitrogen-free fuels. Two sets of experimental results presented in Figure 6 show an increased NO emission under staged combustion conditions. A 17.5K lbm steam/hr watertube unit (11) was modified to enable second stage air to be injected through movable tubes projecting through the front face of the boiler. As burner stoichiometry was decreased NO_x emissions increased from baseline values reaching a maximum at a burner stoichiometry of approximately 80 percent; the increase was greatest when the staged air was injected closest to the burner. A 12K lbm steam/hr firetube boiler (6) was modified for combustion staging by insertion of air delivery tubes from the back of the boiler along the firetube wall to a movable staging ring that directed the secondary air radially inward. Minor reductions in emissions were obtained with natural gas and in most cases emissions increased substantially as the burner equivalence ratio decreased below stoichiometric. In almost all cases for a given staging ring location highest emissions were recorded at the lowest burner stoichiometry.

The increase in NO_x emission with decreasing burner stoichiometry observed in a number of different staging systems suggests the simplified view of staging does not necessarily apply to practical systems. One reasonable suggestion which could account for the observations is based upon the concept that NO formation in gaseous turbulent diffusion flames is strongly dependent upon the rate of entrainment of bulk gases into the reacting mixture. Since reaction zone temperatures are quenched by mixing with the cooler bulk gases, the more vigorous the mixing process, the more rapid is the quench rate, and residence times a high temperature are reduced. This mixing process is controlled to a large extent by the burner momentum which is reduced as the percentage of stoichiometric air at the burner is reduced. The mixing process occurring within the second stage is also important and acts in a similar manner. Consider that the first stage were to be perfectly mixed, and at the location of staged air injection it formed a uniform plug of partially oxidized fuel. The second stage air addition could be considered as two extremes:

1. The second stage air is added and mixes instantaneously producing fully combusted products whose temperature is adiabatic for the reactants.
2. The second stage air is added in finite successive amounts which also mix instantaneously and give adiabatic products. However, as the mixture progressively becomes less rich and reaches stoichiometric proportions, maximum temperatures would occur far in excess of those achieved in (1).

Thus these two extremes represent conditions that would produce high and low NO levels. The real process must be somewhere between these extremes, but they illustrate how the mixing processes that occur in the second stage could influence the final NO level. An alternative hypothesis can be proposed which would also contribute to the observed effects which depends upon oxidation of bound nitrogen specie (HCN, CN, NCO and NH_1) in the second stage. The residence time in the rich primary section should be sufficiently long to ensure minimum exit concentration of these species.

The effectiveness of staged combustion by remote air addition as a technique for reducing fuel NO_x emissions is strongly dependent upon the same system design and operating parameters that dictate thermal NO_x reduction. These are the details of the fuel/air mixing process, particularly the "unmixedness" in the primary zone, the residence time and heat extraction of the primary zone, and the rate of mixing of primary zone products with secondary combustion air.

Evidence also exists that fuel properties are important in determining the effectiveness of staging in controlling NO_x emissions. Thus the control effectiveness is system-dependent and an optimal system configuration developed for one particular combustion device will not necessarily be optimal under different conditions.

The results of several attempts to control fuel NO_x emissions from a number of different combustion devices with remote addition of second stage air is presented in Figure 7. Turner and Siegmund (10) demonstrated substantial NO_x reduction with a 50 HP firetube boiler with remote air addition from a boom protruding into the combustion chamber from the rear of the boiler. These results were obtained at 40 percent of the design firing rate of the boiler due to limitations of the secondary air injection system. Reductions of 50 percent of the total NO_x could only be achieved in a similar laboratory system after extensive modifications were made to the burner (14). Heap et al (5) have reported substantial reductions on the 12K lbm/hr field-operating firetube boiler in which the staged air was introduced from the rear of the boiler through pipes laid along the sidewall. The effect of primary zone residence time could be studied by varying the distance of the staging ring from the burner. At constant primary zone stoichiometry emissions decreased as the distance of the staging injector from the burner increased. Emissions monotonically decreased with primary zone stoichiometry except for the largest primary zone length suggesting an optimum burner stoichiometry for a constant staging configuration, or emphasizing the importance of optimum mixing conditions in the primary zone.

There are practical disadvantages associated with the addition of second stage air via tubes inserted through the boiler walls. A more convenient system would allow the total system to be constructed as part of the burner, thus requiring minor modifications to the boiler. A staging system of this type requires optimization of both the primary zone residence time and the mixing conditions in the primary zone as well as the method of secondary air injection. Cichanowicz and Heap (7) have recently carried out a series of investigations on a laboratory combustor to define these parameters for an optimal staging system for firetube boilers. The resulting burner requires front end modification only and utilizes second stage air addition from the periphery of a refractory primary chamber. The primary zone fuel/air mixing conditions were optimized for the combustion air tangential/axial momentum distribution, fuel atomizer operating conditions, and burner exit geometry.

The individual tests carried out for the evaluation of the influence of these parameters are described in reference (7) and the necessary conditions for optimal staging system performance may be summarized as follows:

- Primary Zone Length - the maximum reduction in NO_x emissions was obtained with maximum mean primary zone residence time and this time was limited by excessive particulate formation.
- Primary Zone Mixedness - conditions which appeared to enhance fuel/air mixing in the primary zone minimized final NO_x levels. The combustion airflow pattern was varied by changing the burner throat diameter, swirl vane design, and burner exit geometry. Fuel atomization input conditions were varied by changing atomizer size (fuel delivery pressure), and atomizing fluid type and quantity. The maximum NO_x reduction was obtained with a high exit velocity, high swirl level, and high atomizing fluid pressure.
- Secondary Air Mixing - staging effectiveness was not strongly dependent upon secondary air injection velocity for the configurations in which air was injected along the firetube wall. NO_x emissions increased significantly when the air was directed radially inward at the exit of the primary zone.

The NO_x emissions produced by the optimal staged burner are shown in Figure 8 as a function of primary zone stoichiometry for three different residual fuel oils, together with the smoke emissions (as Bacharach Smoke Number). Significant reductions in emissions were obtained for all fuels, and further decreases in primary zone stoichiometry were limited by excessive smoke emissions. It is interesting to note that at or above primary zone stoichiometric conditions similar levels of NO_x emissions were measured from the intermediate and high nitrogen fuels.

SECTION 4

CONTROLLED FUEL/AIR MIXING TO LIMIT NO_x EMISSIONS

Air and fuel are supplied separately to the boiler combustion chamber and mixing and heat release take place almost simultaneously. Controlling the rate at which the fuel and air mix can achieve a similar effect as the staged combustion techniques discussed in the previous section - a major portion of the fuel reacts under oxygen-deficient conditions. Even in conventional staged systems the initial fuel/air mixing in the primary region has a substantial effect upon ultimate NO levels. A recent series of laboratory experiments concentrated upon the variation of tangential and axial air momentum flux at the burner throat as a means of controlling fuel/air mixing patterns (7). The burner throat is divided into four concentric channels and the tangential momentum in each channel can be varied independently. A wide range of NO levels were obtained by varying the tangential velocity across the burner throat; however, one general pattern emerged that appeared to apply for all fuels - NO formation was very strongly dependent upon the swirl level in the vicinity of the oil nozzle. This is illustrated by the results presented in Figure 9 which show NO emission as a function of swirl level for two swirl distributions and three fuels. The swirl level relates to increasing the tangential momentum in either the inner or outer channel while the other channel had only axial flow. The total air was divided evenly between the two channels. Thus, the high NO levels were obtained with a tangential momentum distribution which most closely resembed a forced vortex (tangential velocity inversely proportional to burner radius) or an intermediate vortex (tangential velocity constant with radius). The low NO levels were obtained with only axial flow in the channel closest to the nozzle and the tangential velocity profile more closely resembles a free vortex distribution.

The high NO pattern (referred to as Type I) produces a short, high intensity flame which spreads rapidly toward the combustor wall. The low NO configuration (Type II) gives a much longer flame which remains on the combustor axis. Gas concentrations and temperature profiles were measured at one axial location to determine the characteristics of these flame patterns. Three fuels were used: kerosene, kerosene doped with pyridine to give 0.2 percent by weight of nitrogen and a residual fuel oil containing 0.36 percent by weight of nitrogen. Representative results are presented in Figures 10 and 11. The NO/NO_x stack gas emissions without pyridine were 80/90 and 110/115 ppm NO/NO_x for the Type II and Type I flames respectively; with 0.2 percent N as pyridine the values were 160/170 and 215/230 respectively. Despite the similarity in stack gas NO_x emissions, the data shows distinct differences in species profiles that are consistent with the visual differences in flame pattern. CO emissions from the Type II flame are maximum off the centerline in the region where the flame is; the CO peak for the Type I flame is almost on the combustor axis consistent with the observed flame pattern. Without pyridine highest NO_x emissions were measured in the vicinity of the centerline with the Type I pattern. NO_x emissions from the Type II flame are maximum just outside the visible flame region and decrease as the centerline is approached due to fuel rich conditions and low temperatures. The species profiles measured with kerosene doped with pyridine show two distinct differences from the undoped case. High concentrations of NO were detected at radial locations corresponding to the outer edges of the flame. These regions are characterized by excess oxygen and high flame temperatures where fuel NO formation would be maximized. In the low NO flame 35 percent of the peak NO_x level is attributed to NO_2. However, it is most likely that this high apparent NO_2 is due to some conversion of fuel nitrogen compounds in the chemiluminescent NO_2-NO converter.

The results obtained with kerosene can be contrasted with those obtained with a residual fuel oil which are presented in Figure 11. The flue gas NO/NO_x emissions measured for the high and low NO conditions were 550/555 and 260/264 respectively. At this axial distance the NO/NO_x and temperature profiles for the high NO condition are flat which is very different from the profiles measured in the low NO flame. With the low NO mixing pattern fuel apparently concentrates in a region close to axis and mixes with the air relatively slowly reflecting the major differences between the two flames. As indicated in many other experiments, rapid mixing of the fuel and air for a nitrogen-containing fuel maximizes NO production. If fuel NO formation is to be controlled, mixing must be limited to that necessary to promote heat release, but minimize fuel nitrogen conversion in the early stages of the flame. The design of a distributed air addition low NO_x burner will not only require controlled air addition, but the fuel injector must be designed to assist the process by injecting the fuel in such a way that the fuel droplets experience the optimum time/temperature history during their lifetime.

SECTION 5

DISCUSSION

The control of pollutant emissions should not supersede our concern for energy conservation. Thus, a primary factor in the choice of the most appropriate emission control technique for package boilers is its impact upon thermal efficiency. This does not imply that capital cost nor attendant increases in auxiliary plant power requirements can be ignored. However, if the application of some form of pollutant control results in fuel savings, then these savings can be offset against other expenses.

Three control options have been discussed in the previous sections and measurements have been made to assess their impact upon thermal efficiency.

- Fuel Selection. The significant differences in NO_x emissions as a function of fuel type from the two industrial boilers discussed in Section 3 are accompanied by significant differences in thermal efficiency. Figure 12 shows the thermal efficiency as calculated by the ASME Heat Loss Method for both industrial boilers as a function of excess air level for methanol, natural gas, and No. 5 fuel oil. A substantial thermal efficiency penalty is incurred by switching from a high to a low polluting fuel and the use of this technique would require an inexpensive source of alcohol fuel.
- Flue Gas Recirculation. Thermal efficiency is almost independent of FGR as shown in Figure 13 which compares thermal efficiency of the two industrial boilers as a function of the mass of FGR for the three different fuels. Thermal efficiency appears to be far more dependent on fuel type and additional data reported in Ref. (7) shows that with FGR excess air level strongly influences thermal efficiency. The resultant cost-effectiveness of FGR is thus

relatively high for use with natural gas where significant reductions in NO_x can be obtained and relatively low for residual fuel oil because of the ineffectiveness of FGR for nitrogen-containing fuels.

- Combustion Air Staging. Thermal efficiency is relatively independent of the amount of combustion air staging, thus the relative cost-effectiveness is primarily dependent on the control effectiveness or fuel type. Figure 14 shows the thermal efficiency of the firetube boiler as a function of burner stoichiometry for No. 5 fuel oil and natural gas for two overall excess air levels. As with FGR, thermal efficiency is dependent primarily on fuel type and overall excess air level.

Fundamental considerations indicate that flue gas recirculation will not provide adequate control possibilities for those fuels containing significant amounts of bound nitrogen. However, for natural gas-fired package boilers NO_x emissions have been reduced by very low levels (less than 20 ppm) without attendant design problems. At present the major uncertainties associated with the application of flue gas recirculation concern long-term maintenance and reliability and the need for appropriate control equipment. The requirement of dual fuel capability for most industrial equipment and the decreased availability of natural gas necessitates the development of other acceptable control techniques for industrial boilers. From a cost-effectiveness viewpoint it is necessary that alternative control techniques should be developed which are equally effective with natural gas and residual fuel oils.

The potential of staged combustion techniques to control both fuel and thermal NO_x emissions has been appreciated for some time. Considerable success has been reported for utility boilers when the staging process was achieved by biased burner operation or off-stoichiometric firing. The most discouraging feature of all the attempts to stage the combustion process in package boilers has been the back of generality, and therefore, applicability of the results. Also, there can be serious questions raised as to the practicality of the methods of staged air injection. The staging equipment used in the firetube boiler and described earlier was considered as "experimental", and although it withstood the environment within the firetube during the staging investigations, it could not be considered as representative of a practical design.

The possiblity of increasing NO_x emissions from gas-fired systems by attempting to stage the combustion process have been described before, Heap et al (16) described early results using the burner as a staging device in which emissions from a natural gas flame were increased by a factor of three when all the combustion air was not supplied through the primary burner throat. Recent laboratory experiments carried out by De Soete (17) in premixed stratified systems indicate that staging could cause an increase in NO_x emissions. Therefore, the results described in Section are not without precedent.

A low emission oil burner is currently being tested in a field operating firetube boiler. The design of this burner was based upon extensive laboratory tests (7) to optimize the air distribution for staged combustion with a particular fuel nozzle. Reductions in emission levels of greater than 50 percent have been achieved without the need to modify the boiler in any way. Future work should concentrate upon atomizer design since it is known that the liquid spray characteristics have a strong influence upon fuel/air mixing conditions.

REFERENCES

1. Locklin, D.W., et al, "Design Trends and Operating Problems in Combustion Modifications of Industrial Boilers". EPA Report No. EPA 650/1-74-032, April 1974.

2. Barrett, R.E. and Putnam, A.A., "Boiler Emissions - An Inventory of Emissions by Boiler Size and Use". Paper presented at the 69th Annual Meeting of the APCA, Portland, Oregon, June 1976.

3. Mason, H.B. And Shimizu, A.B., "Definition of the Maximum Stationary Source Technology System Program for NO_x Control", Quoted in Ref. 11.

4. Offen, G.R., Kesselring, J.P., Lee, K., Poe, G. and Wolfe, K.J., "Control of Particulate Matter for Oil Burners and Boilers". EPA Report No. 450/3-76-005.

5. Heap, M.P., McComis, C., Tyson, T.J., McMillan, R.E., Zoldak, F.D., Sommerlad, R.E., "Reduction of Nitrogen Oxide Emissions from Package Boilers". EPA Report No. 600/2-77-025.

6. Cichanowicz, J.E., Heap, M.P., McComis, C., McMillan, R.E., Zoldak, F.D., "The Control of Nitrogen Oxide Emissions from Package Boilers", Report in preparation, EPA Contract 68-02-1498.

7. Cichanowicz, J.E. and Heap, M.P., "Design Criteria for Low NO_x, High Efficiency Oil-Fired Firetube Package Boilers", Report in preparation, EPA Contract 68-02-1500.

8. Martin, G.B., "Evaluation of NO_x Emissions Characteristics of Alcohol Fuels for Use in Stationary Combustion Systems". Paper presented at the AIChE Symposium on the Impact of Methanol Fuel on Urban Air Pollution, Boston, Mass., September 1975.

9. Martin, G.B. and Berkau, E.E., "Evaluation of Various Combustion Modification Techniques for Control of Thermal and Fuel-Related Nitrogen Oxide Emissions". Paper presented at the 14th Symposium (International) on Combustion, The Pennsylvania State University, August 21-25, 1972.

10. Turner, D.W. and Siegmund, C.W., "Staged Combustion and Flue Gas Recucle: Potential for Minimizing NO_x from Fuel Oil Combustion". Paper presented at the American Flame Research Committee Meeting, Chicago, Ill., 1972.

11. Cato, G.A., Muzio, L.J. and Shore, P.E., "Field Testing: Application of Combustion Modification to Control Pollutant Emissions from Industrial Boilers, Phase II", EPA Report No. 600/1-76-086a, April 1976.

12. Brown, T.D., Mitchell, E.R. and Lee, G.K., "Low NO_x Combustion: The Effect of External Flue Gas Recirculation on Emissions from Liquid Fuel Combustion". Proceedings of the Combustion Institute European Symposium, September 1973, Academic Press.

13. Shoffstall, D.R., "Burner Design Criteria for Control of NO_x from Natural Gas Combustion: Volume I — Data Analysis and Summary of Conclusions".

14. Muzio, L.J., Wilson, R.P., Jr. and McComis, C., "Package Boiler Flame Modifications for Reducing Nitric Oxide Emissions, Phase II of III", EPA Report R2-73-292b, June 1974.

15. Siegmund, C.W. and Turner, D.W., "NO_x Emissions from Industrial Boilers: Potential Control Methods", ASME Paper No. 73 - IPWR10, 1973.

16. Heap, M.P., Walmsley, R. and Holthuysen, A.M., "The Measurement of Nitric Oxide in Natural Gas and Pulverized Coal Flames, I.F.R.F. Document No. G19/a/2, 1972.

17. De Soete, G.G., Paper presented at the 15th International Combustion Symposium, Tokyo, Japan, August 1974.

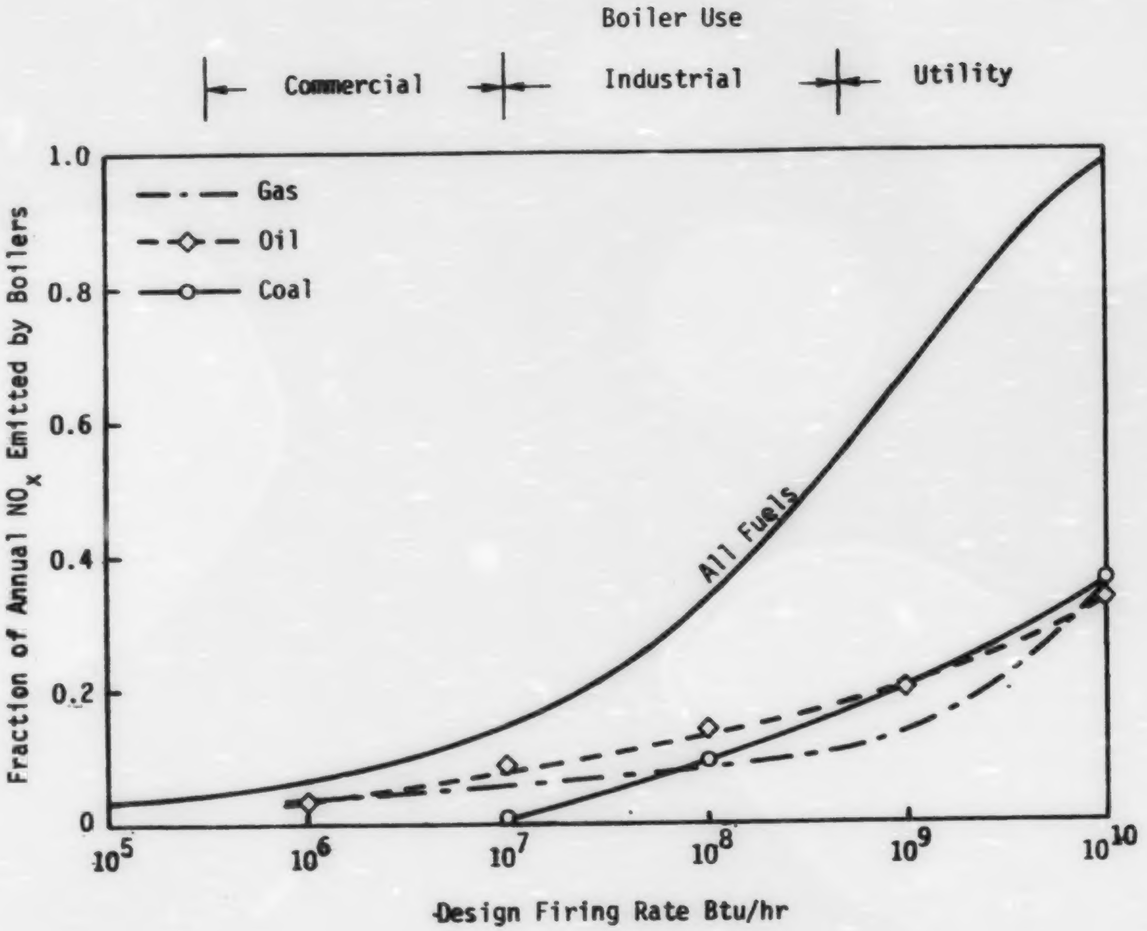

Figure 1. Fraction of NO_x Emitted by Boilers as a Function of Boiler Size and Fuel Capability (after Barrett and Putnam(2))

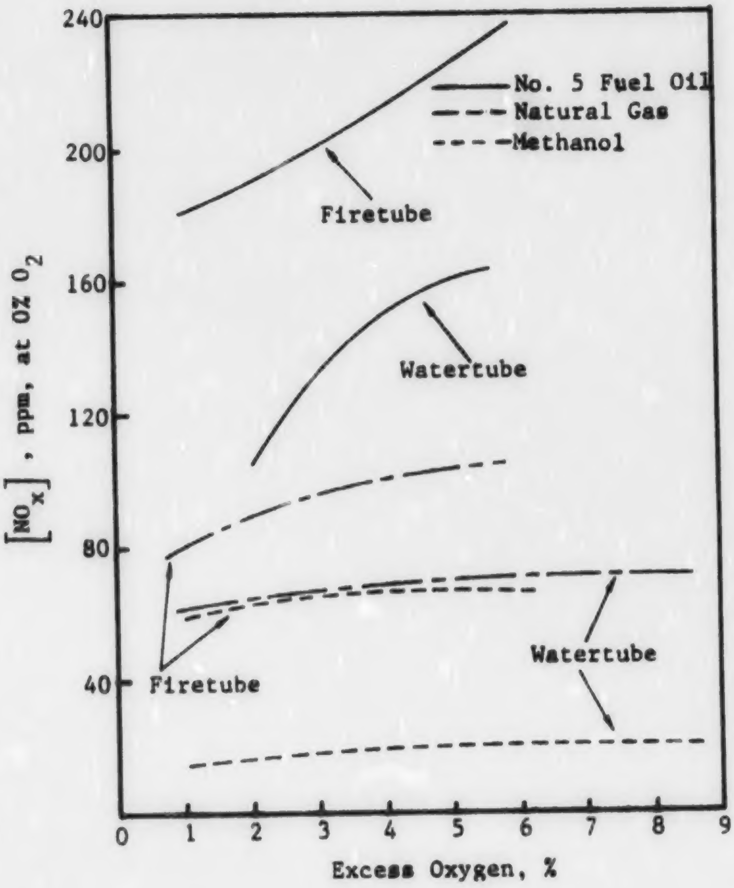

Figure 2. The effect of excess oxygen level on NO_x emissions as a function of fuel type for a firetube and watertube boiler.

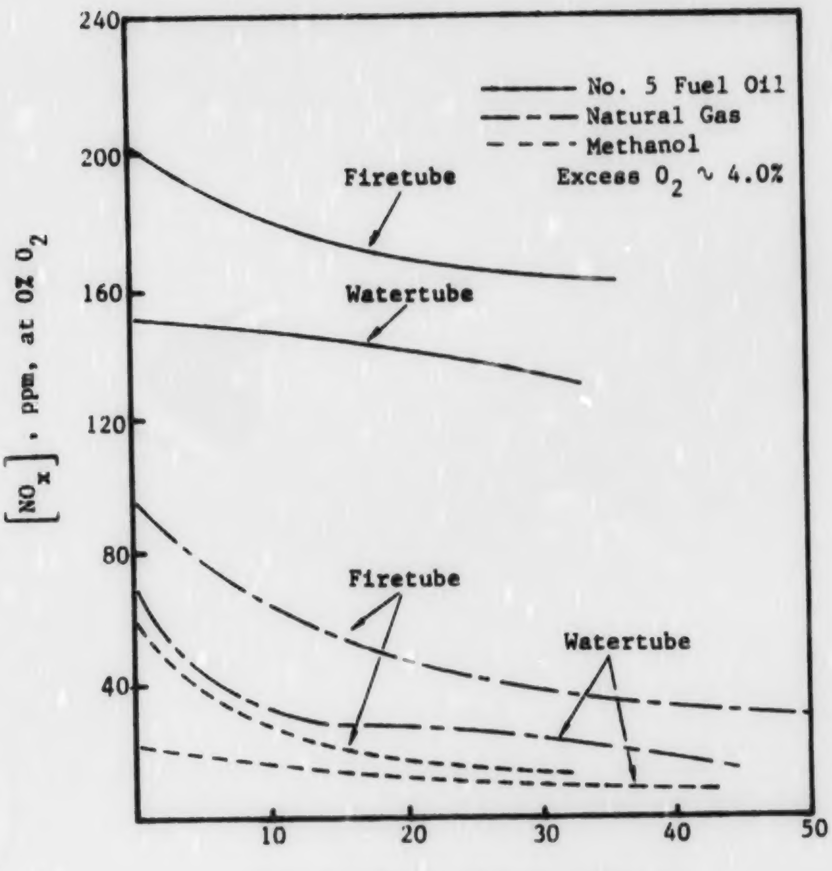

Figure 3. The effect of flue gas recirculation on NO_x emissions from three fuel types from a firetube and watertube boiler.

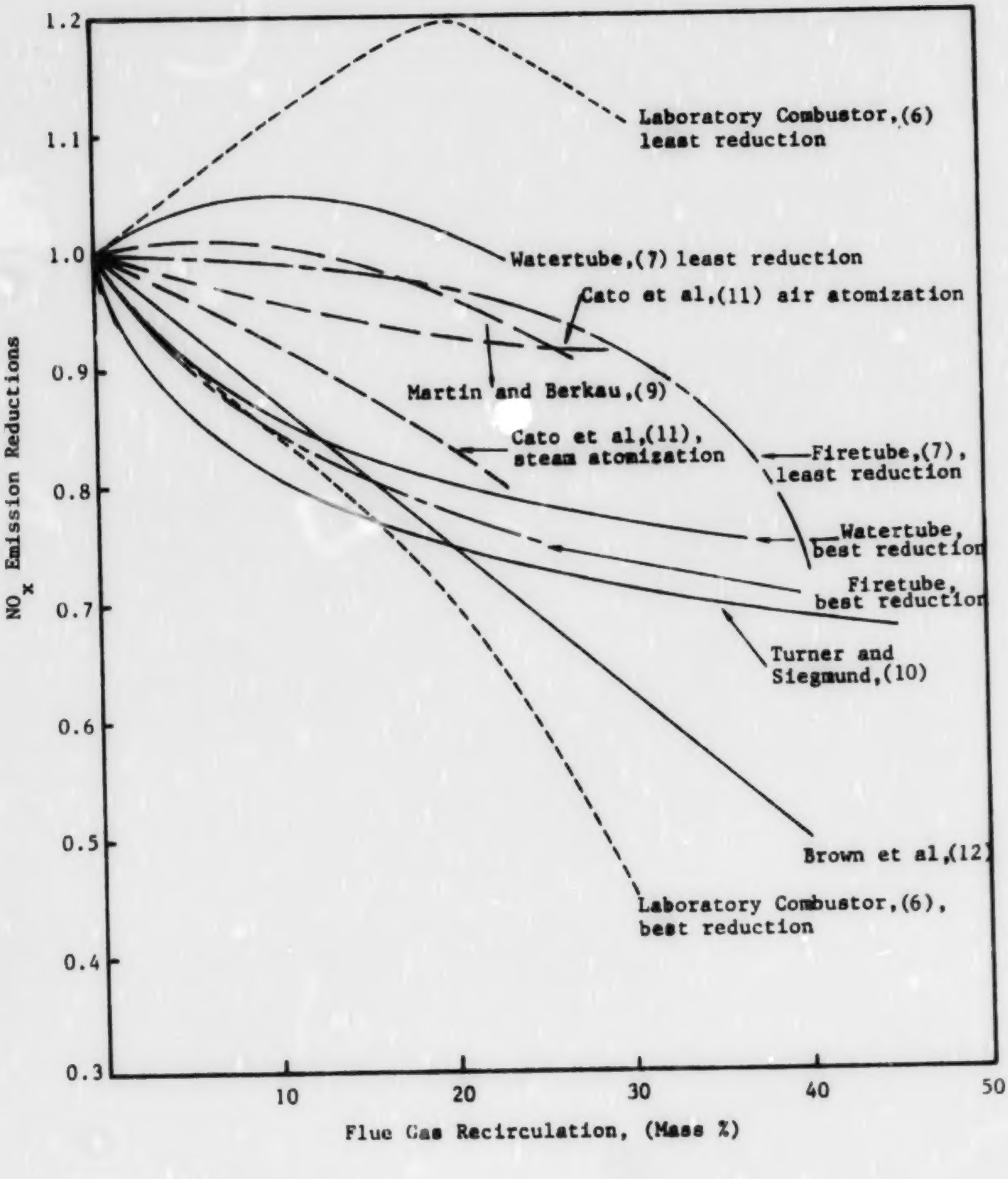

Figure 4. The effect of flue gas recirculation on NO_x emissions from residual fuel oil for a variety of combustion systems.

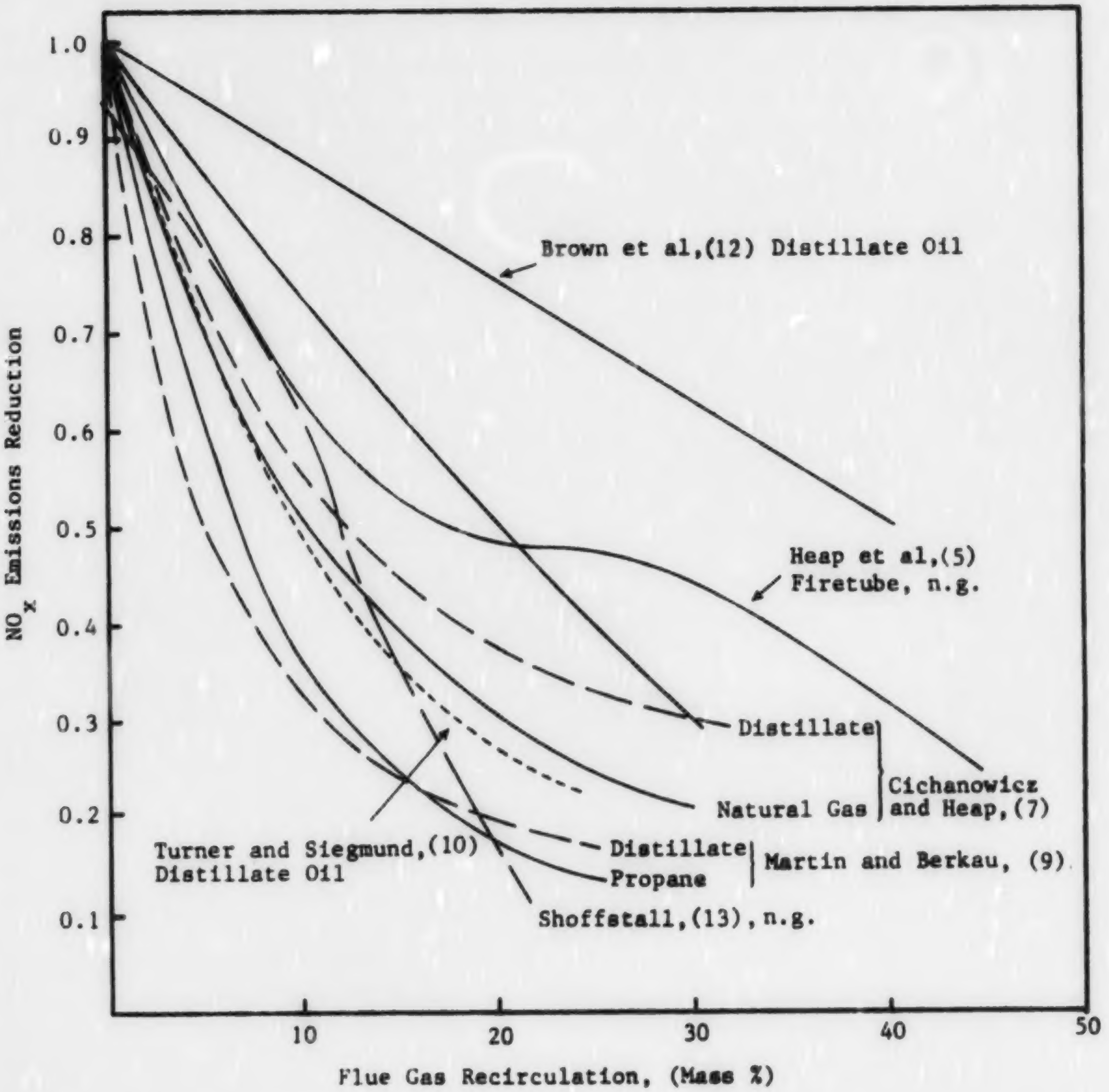

Figure 5. The effect of flue gas recirculation on NO_x emissions from nitrogen-free fuels for a variety of combustion systems.

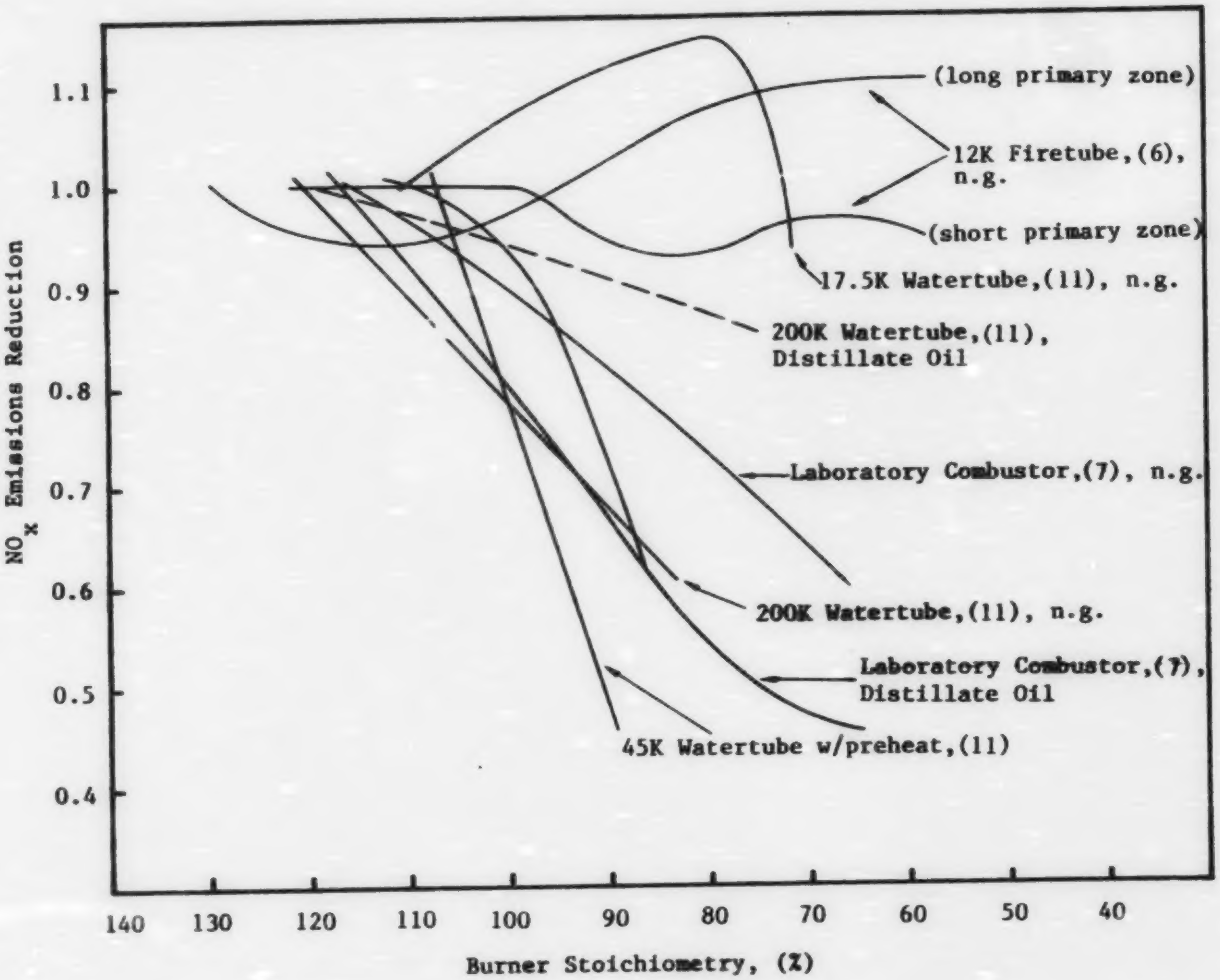

Figure 6. The effect of burner stoichiometry on NO_x emissions reduction for a variety of combustion staging systems.

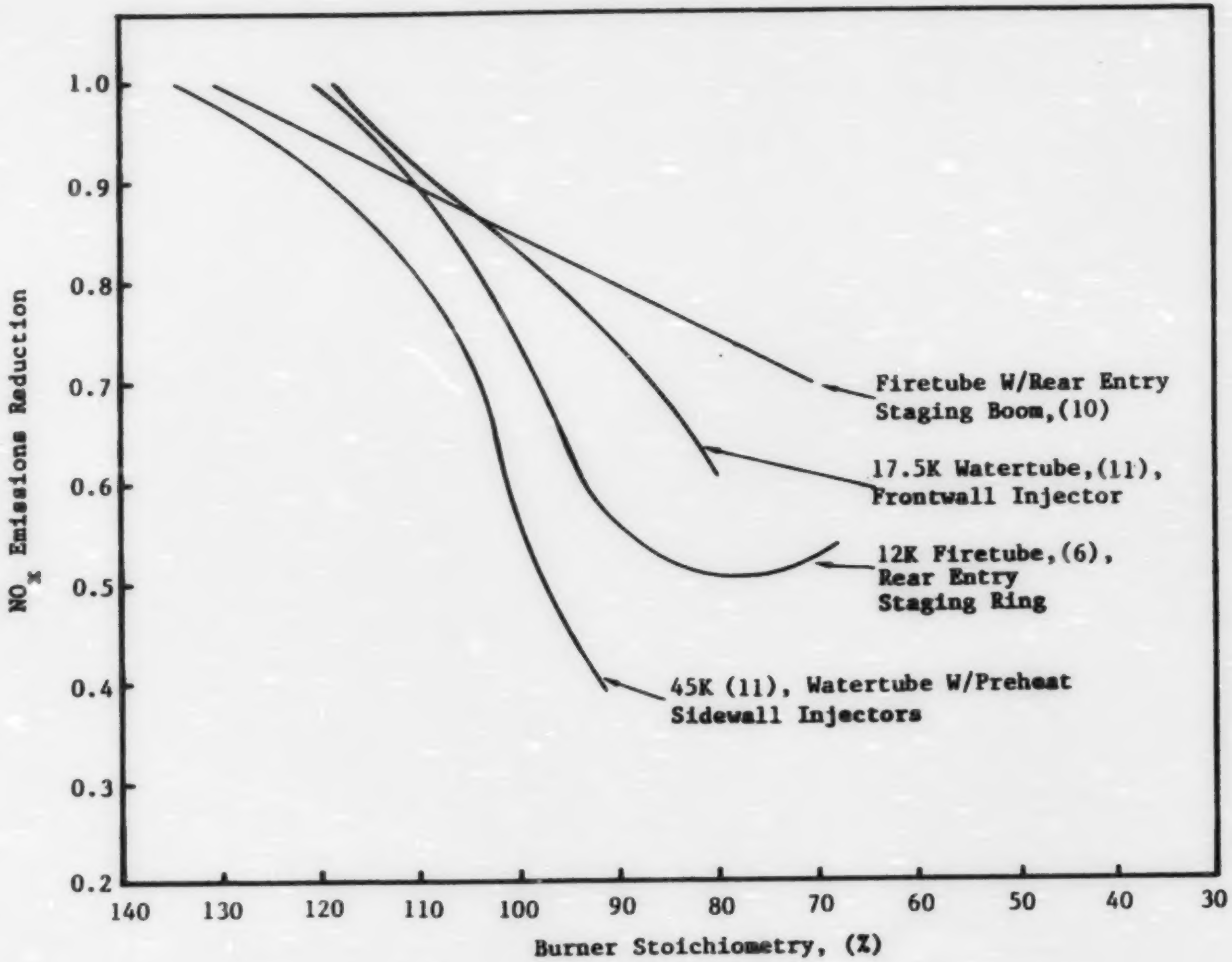

Figure 7. The effect of burner stoichiometry on NO_x emissions reduction from residual fuel oil for a variety of combustion staging systems with nitrogen-free fuels.

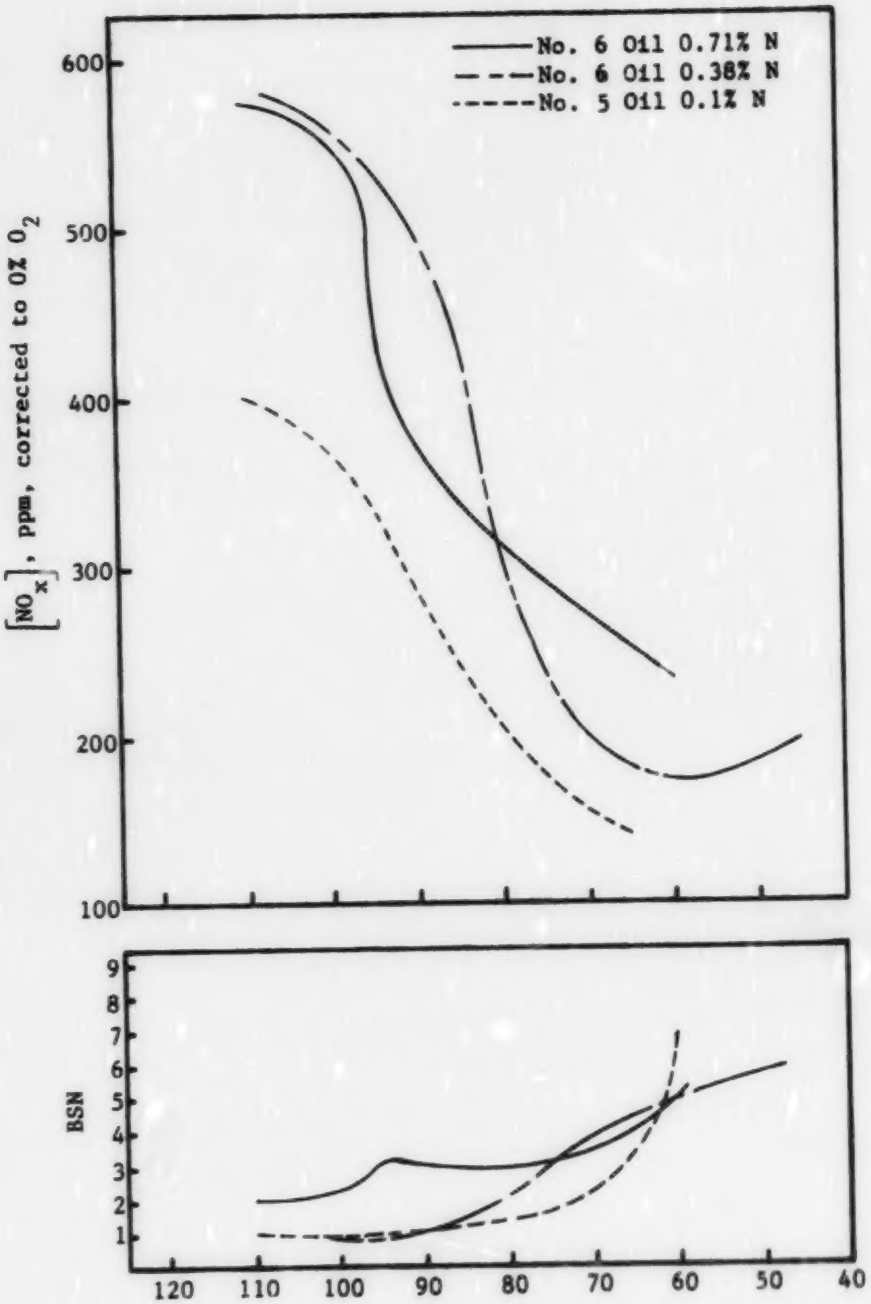

Figure 8. The effect of burner stoichiometry on NO_x and smoke emissions from the refractory staged burner for three residual fuels.

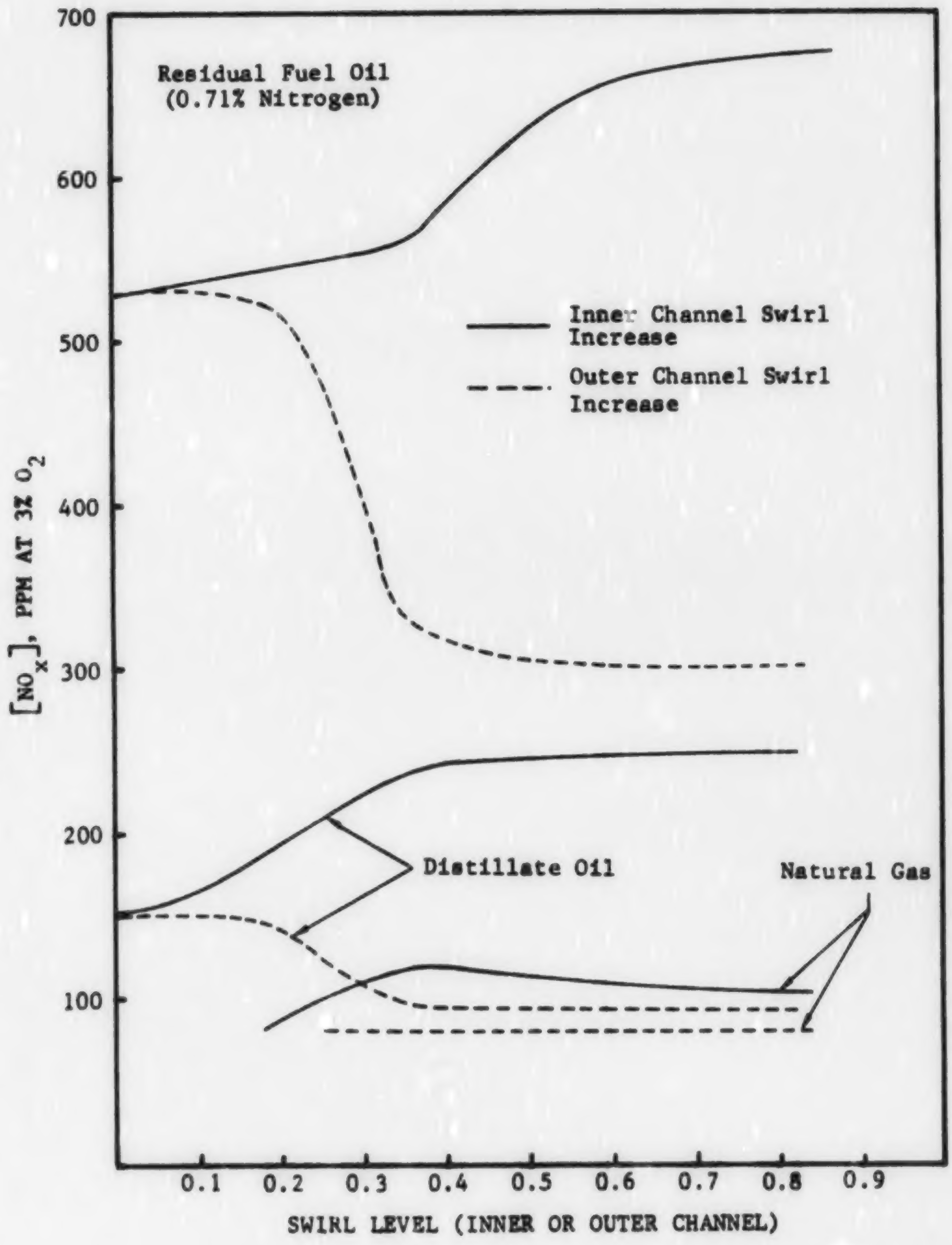

Figure 9. The effect of tangential velocity distribution on NO_x emissions for residual fuel oil, distillate fuel oil and natural gas.

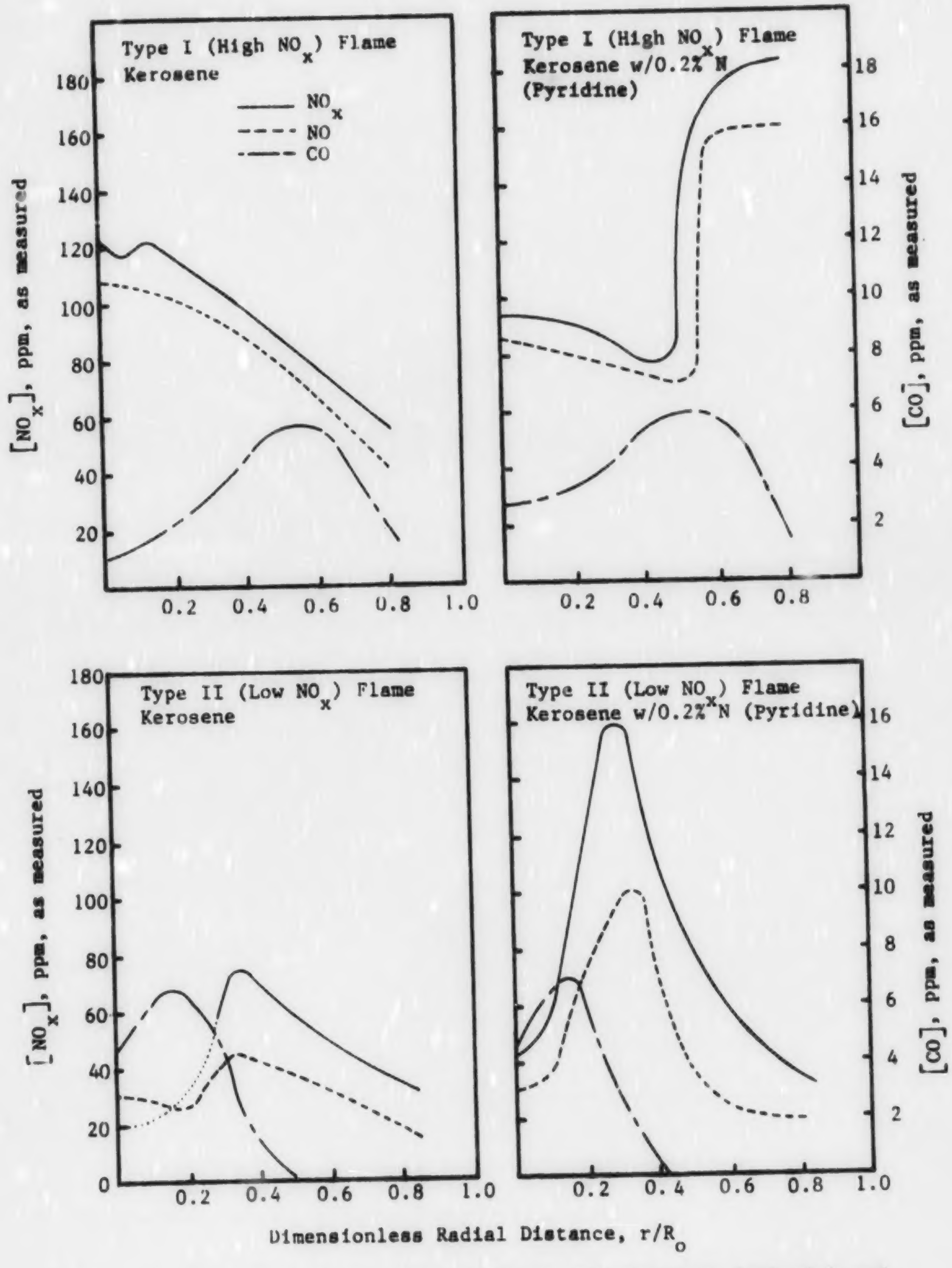

Figure 10. NO$_x$, NO and CO radial profiles with kerosene fuel with and without 0.2% N (as pyridine) for the Type I and II flame patterns.

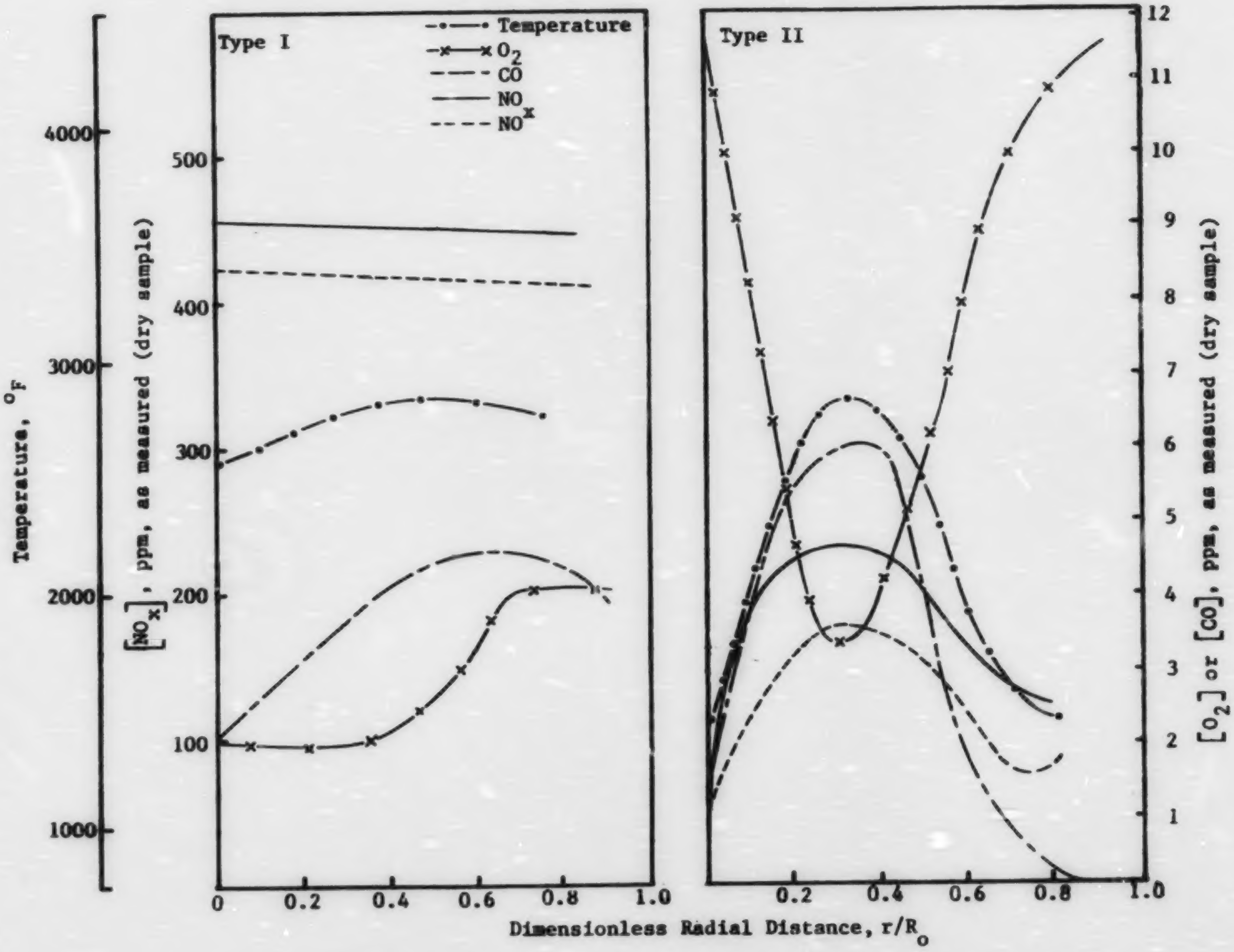

Figure 11. Temperatures (°F) and species profiles (as measured, dry sample) for the Type I and II flame patterns as a function of dimensionless radial distances with residual fuel oil

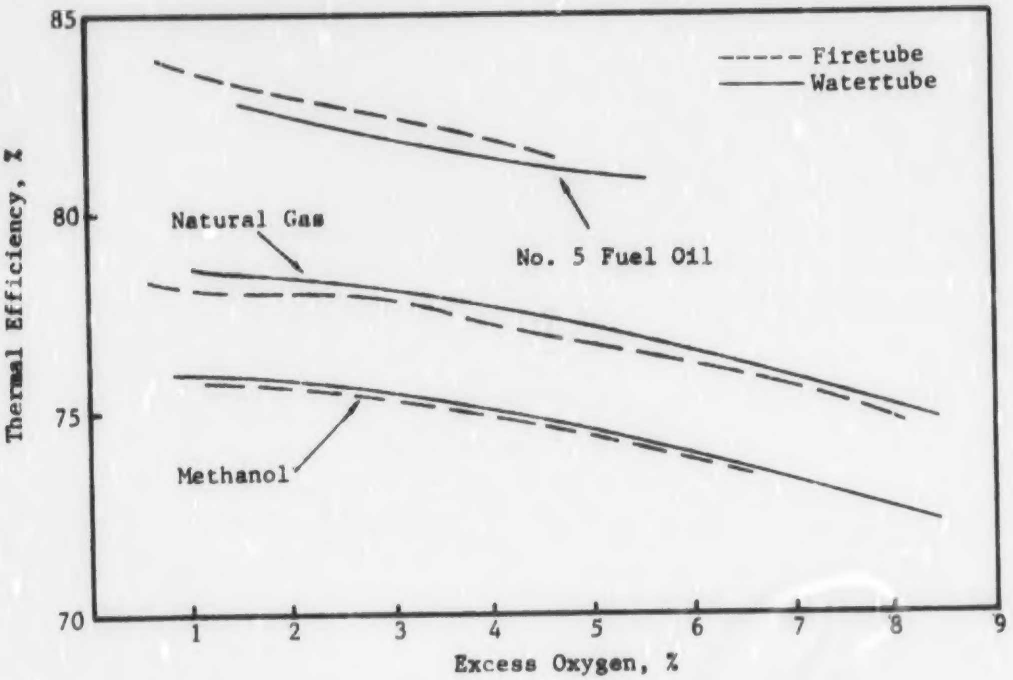

Figure 12. The effect of excess oxygen level on thermal efficiency of the firetube and watertube boilers for three fuel types.

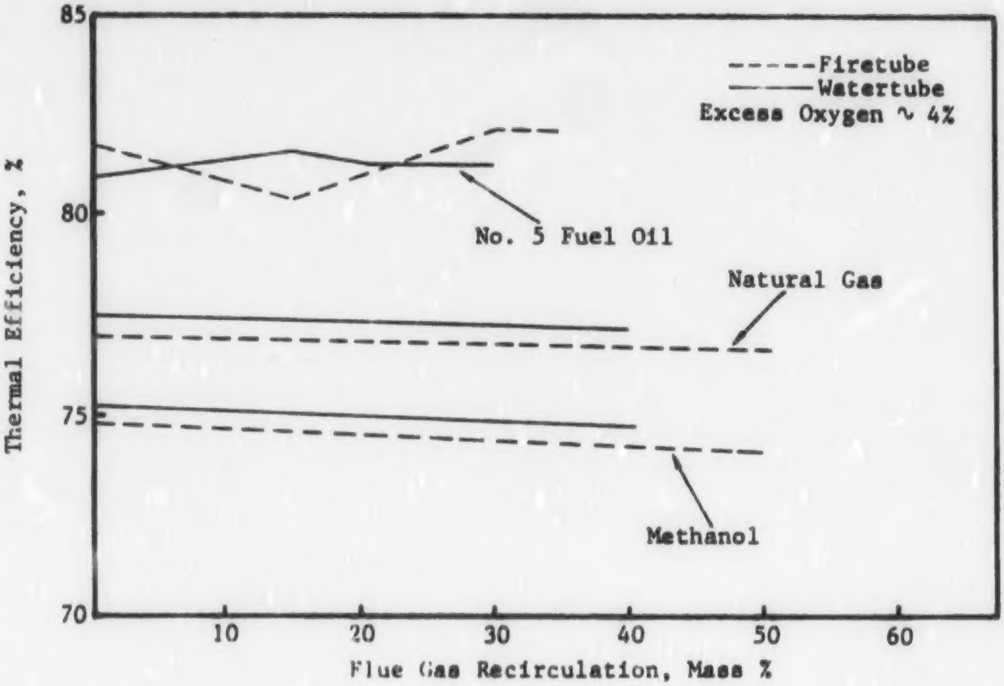

Figure 13. The effect of flue gas recirculation on thermal efficiency of the firetube and watertube boilers for three fuel types.

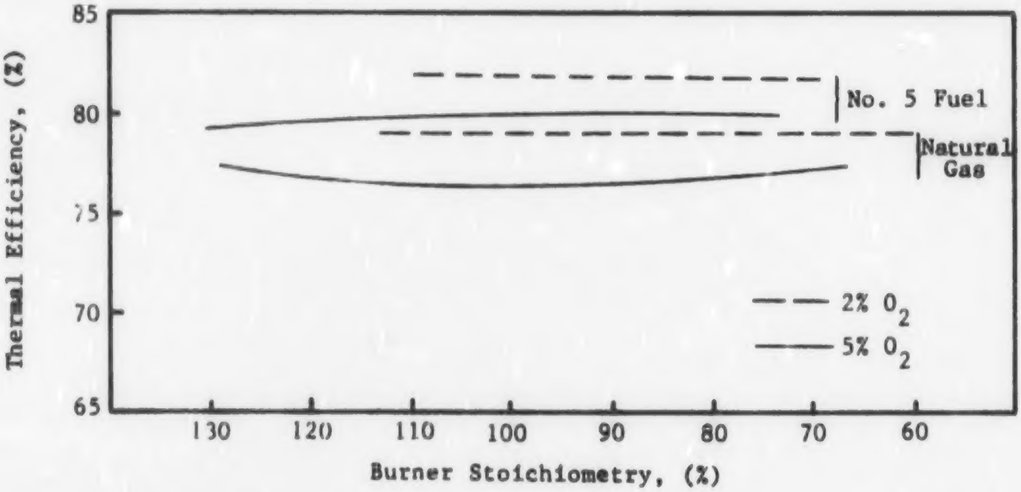

Figure 14. The effect of burner stoichiometry on thermal efficiency of the firetube boiler for two overall excess air levels.

SESSION II:

UTILITY AND LARGE INDUSTRIAL BOILERS

DAVID G. LACHAPELLE
CHAIRMAN

DESIGN AND SCALE-UP OF LOW EMISSION BURNERS FOR INDUSTRIAL AND UTILITY BOILERS

By:

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Energy and Environmental Research Corporation
Santa Ana, California 92705

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SECTION 1

INTRODUCTION

NO_x emissions from coal and oil-fired industrial and utility boilers currently constitute approximately one-quarter of nation-wide NO_x emissions and this fraction can be expected to increase as more emphasis is placed on use of coal for power generation and tighter regulations are imposed on automobile emissions. The EPA has established a broadly-based program to reduce NO_x emissions from large boilers. As part of this program the EPA has established goals for NO_x emission control technology. The goals will require substantial improvement of the state-of-the-art, particularly the 1985 R&D goal of 100 ppm for coal-fired utility boilers.

Low NO_x burner technology capable of meeting these goals has been investigated in pilot scale work but has not been demonstrated on a practical scale. This technology is based on the concept of distributed fuel/air mixing. The basic purpose of the study described here (EPA Contract No. 68-02-1488) is to develop the distributed mixing burner concept to a level where it could be of use to manufacturers and users of large boilers.

Specific objectives of the study are:

1. To design and construct a facility for large-scale burner testing which simulates the geometry and thermal environment of practical boilers;
2. To design and build practical low NO_x burners and test them over a wide range of operating conditions;
3. To investigate application of the low NO_x burner concept to multiburner arrays; and
4. To develop scaling laws to assure that the low NO_x burner concept can be applied to a wide range of furnaces.

SECTION 2

THE DISTRIBUTED MIXING BURNER CONCEPT

To understand the operation of the distributed fuel/air mixing burner, it is necessary to consider the processes of NO formation in fossil fuel flames. NO can be formed either from molecular nitrogen in the combustion air (thermal NO) or from chemically-bound nitrogen in the fuel (fuel NO). Fuel NO can be further divided between NO formed from nitrogen in the volatiles and NO formed from nitrogen in the char.

Uncontrolled pulverized coal combustion in large utility boilers has total NO emissions of up to 1000 ppm. Thermal NO is estimated to represent approximately 15 percent of the total; volatile NO, 65 percent; and char NO, 20 percent. Although volatile NO represents the major portion of the emissions, the other two sources must also be dealt with in order to meet the 100 ppm goal.

In the distributed mixing burner concept the approach to reducing thermal NO is to minimize the peak flame temperature. This method has been utilized successfully in many other applications.

Formation of NO from volatile nitrogen can be minimized by control of flame zone stoichiometry. The key factor is competition between the following two reaction paths:

$$YN + O_x \rightarrow NO + \ldots$$

$$YN + ZN \rightarrow N_2 + \ldots$$

with the NO path dominating under lean conditions and the N_2 path dominating under rich conditions.

In turbulent diffusion flames, although the stoichiometry may be overall rich, there will be local lean zones which will result in some NO formation.

If residence time in the rich zone is adequate, however, any such NO can be reduced by hydrocarbon fragments:

$$NO + CH \rightarrow YN + \ldots$$

NO may also be reduced by heterogeneous reduction by char.

In the case of char NO, no effective control methods are available. Instead the approach used here is to maximize evolution of nitrogen with the volatiles. Laboratory work has indicated that most of the nitrogen can be driven out of the char by providing for adequate time at high temperature.

Figure 1 illustrates the distributed fuel/air mixing concept. The coal and primary air are injected with a moderate axial component. This stream is surrounded by a secondary air stream with a swirl component for stabilization. Tertiary air for burnout is added axially around the periphery of the burner. This arrangement results in a weak shear layer between primary and secondary streams which leads to gradual mixing. This provides for a hot, rich recirculation zone with adequate time at high temperature to maximize evolution of nitrogen from the char and adequate time in the rich zone to reduce most of the NO that may be formed.

Axial addition of the tertiary air leads to a large flame zone and thus provides for heat extraction prior to completion of burnout. This, along with dilution of the tertiary air by combustion products, lowers the peak flame temperature and thus reduces thermal NO.

Burners of this type have given excellent results in pilot scale testing. An example of the pilot data is shown in Figure 2. It can be seen that levels as low as 100 ppm were achieved. Figure 2 also illustrates the importance of coal injector design. (The coal spreader injector identified in Figure 2 gives a flow pattern similar to that shown in Figure 1.)

SECTION 3

PROGRAM DESCRIPTION

The current program has been set up to demonstrate the applicability of the distributed mixing concept to practical systems. Specific questions which must be addressed include:

1. Effects of scale and furnace geometry;
2. Effects of burner interactions in multiburner arrays;
3. Impact of larger flame zones on furnace capacity;
4. Potential corrosion problems due to larger rich reaction zones; and
5. Potential slagging problems due to reduced ash softening temperatures in the rich zones.

The test facility and experimental plan have been designed to provide answers to these questions.

TEST FACILITY

The main features of the burner development test facility are as follows:

- Two boiler simulators
  - -- a utility boiler simulator capable of firing at 120 x 10^6 Btu/hr.
  - -- a watertube package boiler simulator capable of firing at 12 x 10^6 Btu/hr.
- Capability to fire pulverized coal, residual oil, or simulated low Btu gas in both units. (The simulated low Btu gas is produced by partially oxidizing propane and adding appropriate quantities of nitrogenous species.)

- Sheet steel walls cooled by water sprays. The water sprays are divided into individually controlled sections.
- Variable geometry for the large utility boiler simulator.
- Air preheat capability to 800°F.
- Independent control and measurement of numerous air streams.
- An on-line pulverizer capable of producing particle size distributions similar to those in industrial practice.
- A steam source for oil atomization.
- Flexible burner positioning and probe access. Flexibility of probe access is important for detection of reducing conditions near the walls that could lead to excessive corrosion.

Variable geometry for the utility boiler simulator is achieved by means of a movable back wall. Figure 3 illustrates the range of positions of the movable wall which can provide for variation of furnace volume by up to a factor of 3. Figure 3 also shows the range of firebox residence times that can be obtained and compares them with existing boilers that have been tested for NO_x reduction.

Construction of the facility is essentially complete and testing has recently started on a 50 x 10^6 Btu/hr burner.

TEST PROGRAM

The first and most important element of the test program is the development of a large-scale distributed mixing burner capable of operating with low emissions and high efficiency under conditions typical of practical boilers.

Other key elements of the test program include:

- Testing of three burner sizes and three furnace volumes to generate data for development of scaling procedures;
- Extension of the distributed mixing concept to multiburner arrays including investigations of burner spacing and burner-to-burner variations in stoichiometry;

- Firing of low Btu gas, both alone and in combination with coal; and
- Detailed flame probing to provide insight into the NO formation process.

SECTION 4

TEST RESULTS

Testing of the burner illustrated in Figure 4 has recently commenced. The burner has an annular coal injector which provides a flow pattern similar to that shown in Figure 1. Secondary air swirl levels are variable and tertiary air is introduced through four axial injectors.

The first test results obtained are compared with results from the pilot tests for similar operating conditions in Figure 5. (Primary zone stoichiometry is defined as the fuel/air equivalence ratio based on the sum of the primary and secondary air flows.) The fuel in the large-scale test was Utah high-volatile bituminous with 1.24 percent N and 0.6 percent S. The large-scale results are seen to compare favorably with the pilot scale results.

SECTION 5

CONCLUSIONS

The first set of test results shown in Figure 5 is very encouraging and at this point it appears that the distributed mixing burner has a very promising future.

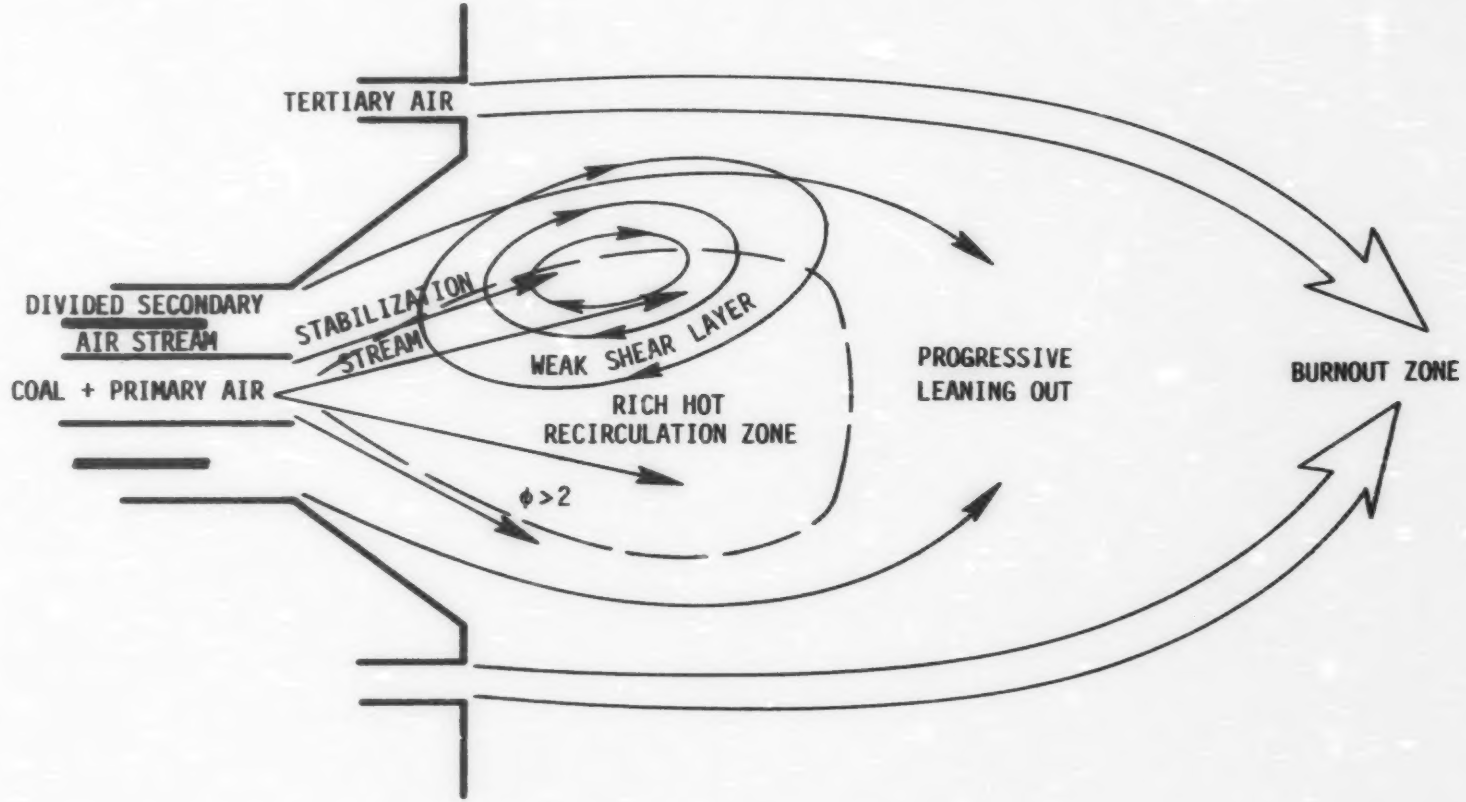
TERTIARY AIR
DIVIDED SECONDARY
AIR STREAM
COAL + PRIMARY AIR
STABILIZATION
STREAM
WEAK SHEAR LAYER
RICH HOT
RECIRCULATION ZONE
$\phi > 2$
PROGRESSIVE
LEANING OUT
BURNOUT ZONE

Figure 1. Distributed Fuel/Air Mixing Concept

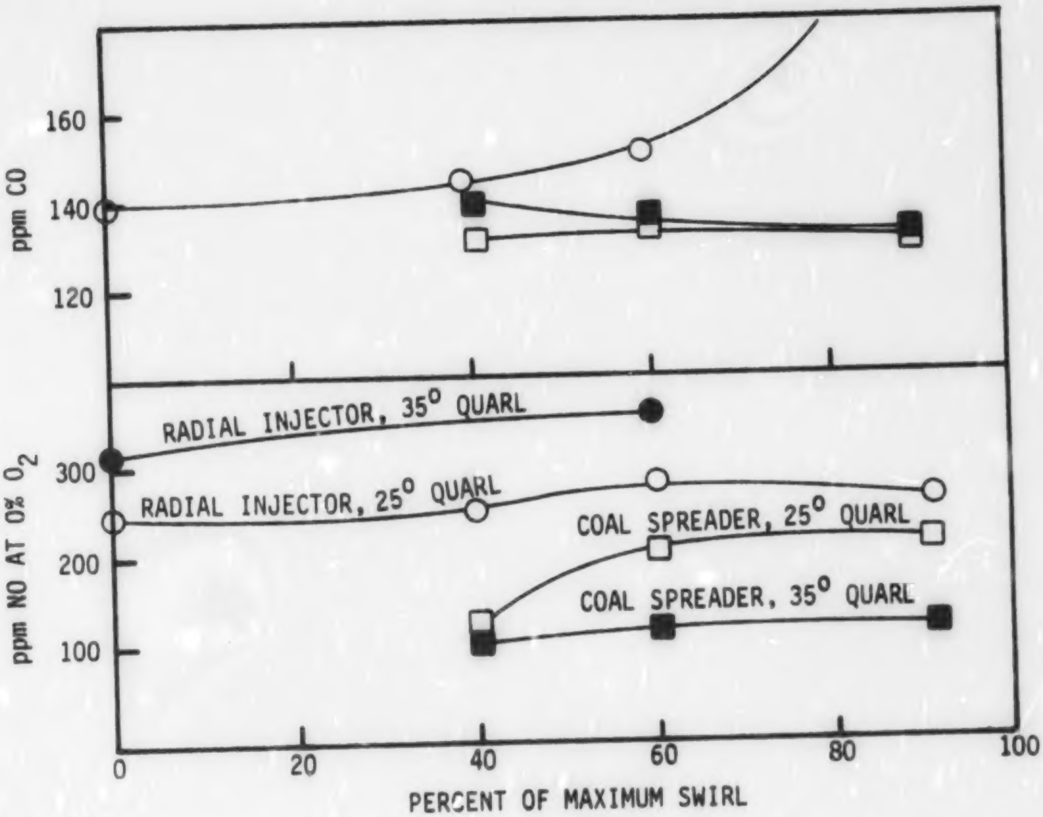

Figure 2. Pilot Scale Test Results

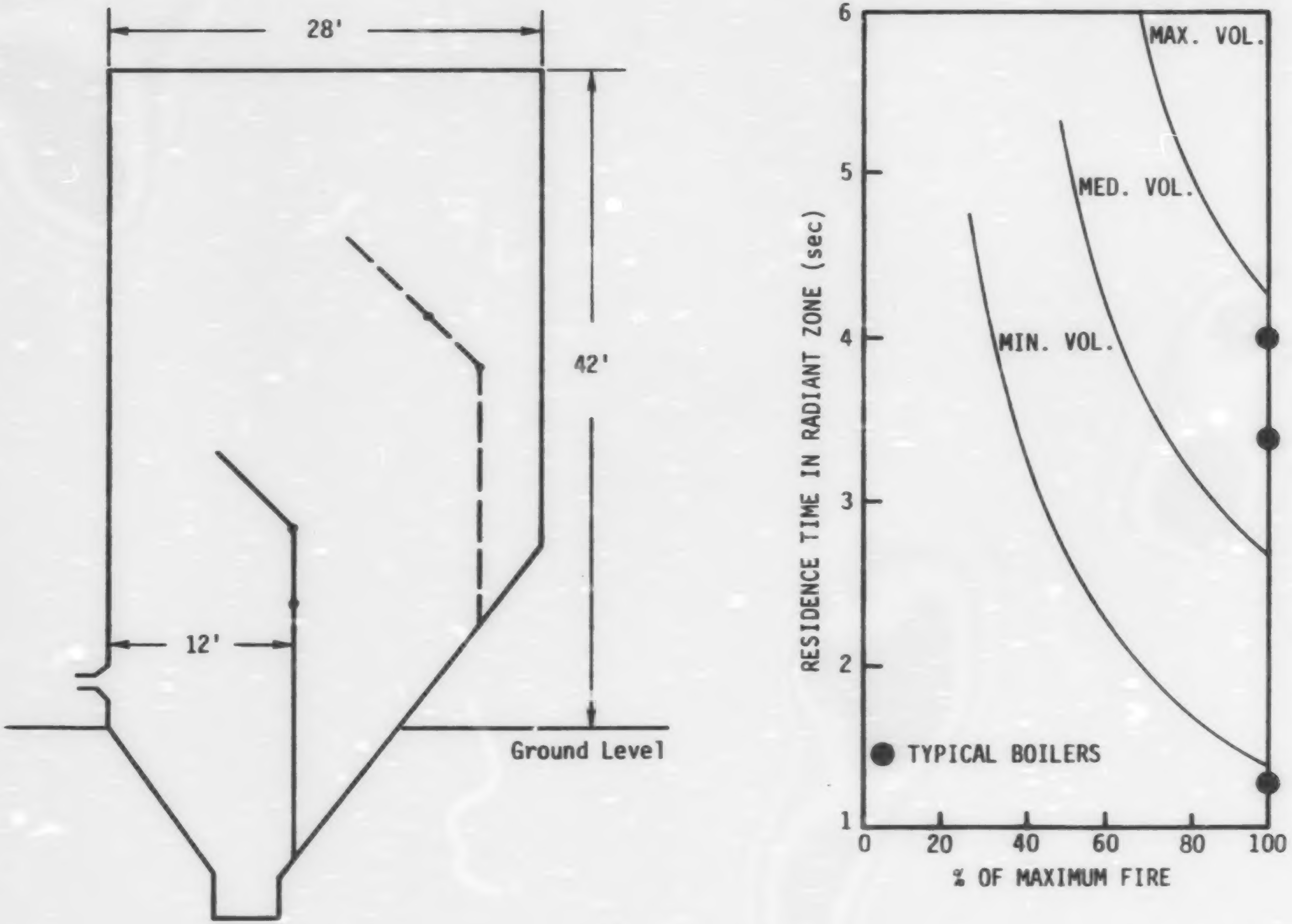
28'
42'
12'
Ground Level
RESIDENCE TIME IN RADIANT ZONE (sec)
1
2
3
4
5
6
0
20
40
60
80
100
% OF MAXIMUM FIRE
MAX. VOL.
MED. VOL.
MIN. VOL.
● TYPICAL BOILERS

Figure 3. Variable Geometry Combustor

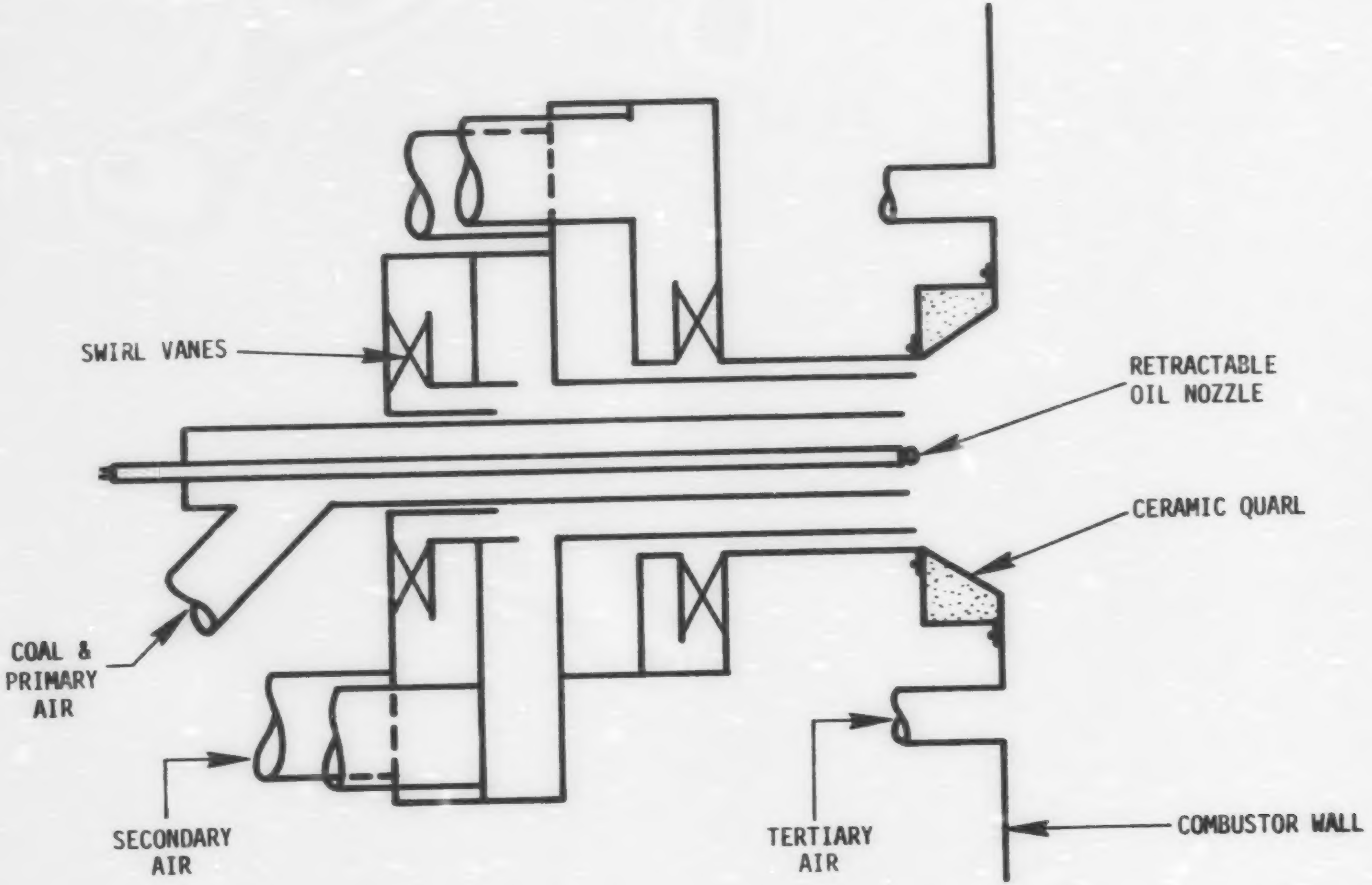
SWIRL VANES
RETRACTABLE OIL NOZZLE
CERAMIC QUARL
COAL & PRIMARY AIR
SECONDARY AIR
TERTIARY AIR
COMBUSTOR WALL

Figure 4. 50 x 10^6 Btu/Hr Distributed Mixing Burner

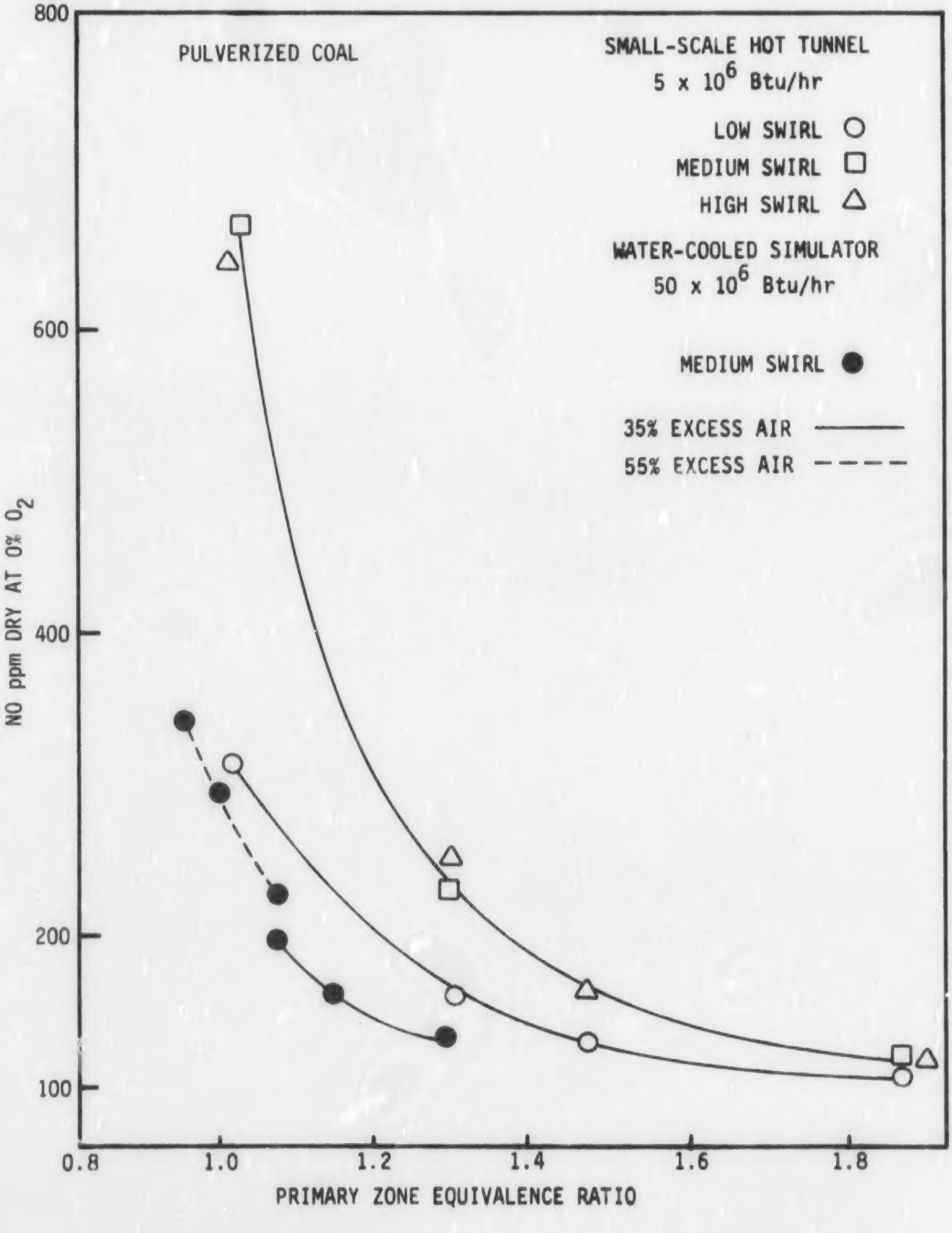

Figure 5. Test Results

STATISTICAL CONSIDERATIONS IMPORTANT IN ANALYZING MEASURED CHANGES IN WATER-WALL TUBE THICKNESS

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ABSTRACT

Most measurements of tube thickness are made ultrasonically. Present ultrasonic thickness gauges require calibration. Their calibrations are not much more accurate than their measurements. Statistical analysis of data for significance or confidence requires identification of portions of the data deserving of being treated as having uncorrelated variations/fluctuations/errors. When calibration variations are comparable with measurement variations this requires that each such portion consist of one or more recalibrations (actual or potential) together with the measurements to which those calibrations apply. Placing in separate portions individual measurements which share a calibration can, under these circumstances, greatly bias the calculation of the apparent certainty of the statistical analysis (usually indicating falsely high precision).

If the practice of calibrating once for each row of measurements on a given wall is followed, the natural portion is a summary — median, mean or rob/res estimate — of the thickness changes found for that row. Estimates of error must then depend on variation from row to row. Using a conventional sum of squared deviations among rows as a basis for significance or confidence can lead to unduly wide allowance for uncertainty, because real trends — say from top to bottom of a wall — are likely. The use of mean square successive differences, introduced more than a century ago by Erastus deForest, and used for eight decades in artillery statistics, can do much to obviate this difficulty. (The needed tables are available.)

Boilers tend to have two furnaces; furnaces tend to have four water-walls; observed numbers are such that results for upper/lower half walls often seem interesting. Together or separately, these facts make parallel significance tests or parallel confidence intervals almost inevitable. Good engineering/scientific practice then leads any/all of us to focus our attention on the most striking results. This raises problems of multiplicity, of simultaneous confidence or of parallel significance. The simplest approach to these problems, sometimes associated with the name of Bonferroni, is easy to understand and use. (More recondite approaches, when appropriate, will yield slightly tighter results.)

Prepared in part (1) in consequence of work at Princeton University on the statistical analysis of corrosion data on the Widow's Creek No. 8 boiler, and (2) in connection with research at Princeton University sponsored by the Army Research Office (Durham).

A. Calibration Practices And Statistical Analysis

1. No Instrument Can Be Trusted Without Calibration

In considering the statistical analysis of measurement data, it is too easy, often fatally easy, to forget that no instrument is automatically perfect, that every instrument requires calibration at one time or another.

Many measuring devices are easily calibrated to much greater precision and accuracy than is required of them in their daily use. Familiarity with such examples, which are not as common as might be naively thought, can seduce us into treating all instrumental measurement as only subject to the sort of "errors of measurement" that are revealed by some relatively simple form of repeated measurement. In many areas, however, ranging from the measurement of total atmospheric ozone by Dobson meters to the measurement of changes in tube thickness by ultrasonic gauges, calibration is made with a precision/accuracy only comparable with that demanded of the measurement.

2. Simple Algebra Of Variances

If σ_c^2 is the variance of a calibration, σ_m^2 the variance of a measurement, and if we make a single calibration and then k measurements, all dependent upon the single calibration, but otherwise independent, the variance of the mean of the measurements will be

$$\sigma_c^2 + \frac{1}{k}\sigma_m^2$$

If these measurements are repeated measurements of the same quantity, and are, in fact, independent, then the sum of their squared deviations from their mean will average (we shall shortly write $\doteq$ for "equals on the average")

$$(k-1)\sigma_m^2$$

(There will be NO contribution from σ_c^2 since we are dealing with differences all of which share a common calibration.)

Thus if we look at the sum of squared deviations divided by $k(k-1)$ we have

$$\frac{\Sigma(y-\bar{y})^2}{k(k-1)} = \frac{s^2}{k} \doteq \frac{1}{k}\sigma_m^2$$

which, were $\sigma_c^2 = 0$, would be an estimate of the variance of our mean of k measurements sharing a common calibration.

3. Consequences

But if $\sigma_c^2 \neq 0$, as almost always happens, then the average value of s^2/k will be TOO SMALL, by comparison with the variance of our mean. If we use it as if it were an estimate of this variance we will make TOO SMALL an allowance for the variability of our mean — possibly greatly too small. So we must find some other way to assess the variability that concerns us.

This is but a single instance of a broad principle: differences between measurements that share a common calibration, cannot serve to honestly assess the variability of a mean (or other combination) of these values.

If we are to assess the variability of an extended mean, for example a mean thickness for measurements spread, in levels, over the whole of a boiler wall, we must base our assessment on differences that combine differences in calibrations *in the proper ratio* to differences of measurements.

4. Proper Grouping And Its Consequences

The simplest way, and often the only reasonable way, to do this is to divide the measurements of our extended mean into groups in such a way that those measurements that share a calibration fall into the same groups. Indeed, it is often simplest to let each group consist of just those observations that share the same calibration. The variability of the extended mean, which is now the mean, perhaps weighted, over all groups of the means within the individual groups, is now to be judged in terms of the differences from one group mean to another.

By doing this we have

(1) not changed the value of the extended mean whose precision it is our purpose to assess,

(2) ensured that the variability due to (always imperfect) calibration will be properly represented,

doing this by telling the truth as to what observations, or combination of observations are subject to approximately independent variations — variations of the whole measurement process, including calibration.

What price may we have paid for this honesty about calibration? We have

(3) left ourselves with fewer values, now the group means, upon whose differences we are to base our estimate of the stability of the grand mean. As a consequence our statistical computations will involve fewer "degrees of freedom" and our allowances for uncertainty will be slightly larger (10% increase in length of confidence intervals will be rather common).

(4) left ourselves with differences, those between groups, which include true differences in what is measured, say between top and bottom of a wall, as well as in the measurement process.

Of these two, the first is inevitable. So long as calibration makes a substantial contribution to overall measurement error, we cannot have appreciably more degrees of freedom associated with a single extended mean than one less than the number of calibrations.

The second is only inevitable in part. Our next act is to learn how to reduce its impact on our calculations.

All the considerations of this section apply equally well to changes, where, for example, the pattern of 1 calibration and k measurements occurs both "before" and "after" for each level, and the mean change for each level is the basis for our further analysis.

B. Successive Differences Can Ease Our Problem

5. The Prototypical Problem

We now face a problem, which the exigencies of practical measurement are very likely to put into the following shape:

- measurements are taken in levels across a boiler wall, with each level of measurements sharing a calibration
- there are likely to be trends in the quantity measured, apparent change in thickness
- differences between observed row means, while reflecting both calibrations and measurements correctly, also reflect differences in this trend.

This is of course a classical problem, one that arises in many situations. One simple and classical case arises in judging the precision (not accuracy) of artillery fire. When a gun is fired repeatedly under as carefully constant circumstances as possible, there will remain trends due to (a) temperature changes for the gun and the powder, (b) changes in the wind, and (c) perhaps other relevant circumstances. If the differences in point of impact — corresponding to our level means — are to be turned into an assessment of imprecision, we will do well to minimize the effect of trends upon our assessment.

6. Alternative Solutions

Two sorts of approach to this problem have been used, and a third sort is possible. The most rigid approach is to choose a functional form — linear, quadratic, or what have you — and act as if we knew that the trend had this form, fitting the unknown constants, subtracting the fitted trend from the observations to form residuals, using these residuals as a basis for assessing precision. All the necessary procedures are classical (though modern robust/resistant techniques may offer further advantages). Experience with many fields suggests that this approach is often far too rigid.

The second approach is beginning its second century. Erastus deForest, a Yale professor, suggested in 1874 using only differences between (physically) adjacent observations (sometimes simple differences, sometimes higher differences) as a basis for assessing the contributions to variability associated with the observations (falls of shot, level means, etc.) themselves. This procedure is quite flexible, relying upon the *local* properties of the trend, rather than on any assumed overall shape. It has been used in ballistics (e.g., artillery precision) for more than 80 years. It has not been used widely enough, though there is sufficient experience to be sure that it is very often very helpful.

A third class of approach, not yet investigated, would be to apply modern (robust/resistant) techniques of smoothing, either to the observations themselves or to differences of adjacent observations, and then to work with the values of the *rough*, where

rough = observed MINUS smooth

For the present, we do not have the knowledge needed to judge the effectiveness of such an approach.

Accordingly it is natural to plan to use the second approach. What do we need to implement its use?

7. Mean Square Successive Differences-Calculation

Exhibit 1 shows the basic calculations for the apparent changes in tube thickness in the side wall of the B furnace of the Widow's Creek No. 5 boiler. We have chosen to summarize either set of successive differences by the sum of its squares.

Suppose we choose to use the 1st differences, how are we to use the sum, 1199.89, of their squares? First, we need a divisor to convert this sum to an estimate of the contribution of overall measurement variability to the variability of a level mean. Once we have this, dividing by the number of level means will give us an estimate of the variability of the extended mean — the grand mean of level means.

8. Mean Square Successive Differences-Denominators

The first of these problems was solved by deForest, and independently by many other later writers. The appropriate divisor is always the product of

- the number of squares summed
- the sum of squares of the coefficients when the difference is written out in detail.

For the first few orders of difference we have the following coefficients and sums of squares:

$$\begin{aligned} \text{First } (y_{i+1}-y_i)&:\ +1,\ -1;\ 1^2+1^2=2 \\ \text{Second } (y_{i+2}-2y_{i+1}+y_i)&:\ 1,-2,1;\ 1^2+2^2+1^2=6 \\ \text{Third } (y_{i+3}-3y_{i+2}+3y_{i+1}-y_i)&:\ 1,-3,3,-1;\ 1^2+3^2+3^2+1^2=20 \\ \text{Fourth}&:\ 1^2+(-4)^2+6^2+(-4)^2+1=70 \end{aligned}$$

Thus if we wished to base an estimate of overall measurement variability (more or less increased by contributions from the trend) on the information in the individual columns of Exhibit 1 we would obtain the following alternative results:

$$\text{(level means)}\quad 206.72=\frac{2597.81-\frac{1}{11}(-76.43)^2}{11-1}$$

$$\text{(first diffs)}\quad 59.99=\frac{1199.89}{20}\quad \text{where } 20=(10)(2)$$

$$\text{(second diffs)}\quad 59.76=\frac{3226.82}{54}\quad \text{where } 54=(9)(6)$$

The agreement between the latter two is shockingly — and accidentally — close. The real point is that both of the values are less than 30% of the value based on the level means themselves. The only appropriate conclusion is that basing our estimate of variability of the wall mean on

$$\frac{206.72}{11}$$

rather than on

$$\frac{59.99}{11}\quad \text{or}\quad \frac{59.76}{11}$$

involves an unacceptably large, quite inappropriate contribution from trends up or down the wall.

9. The Widow's Creek Experience

The example just discussed is the most extreme of the eight arising in the analysis of the Widow's Creek No. 5 data. Exhibit 2 shows the alternative estimates of overall measurement variability for all eight walls.

Which estimates of overall measurement variance for level means to take in practice has to be a matter of judgement. If we have no adequate basis for final choice, we should choose the lower of the two orders of differences involved, for reasons to be explained below.

The writer would consider any of the following choices clearly defensible for the set of data illustrated:

Exhibit 2

Alternative Estimates of Overall Variance of Measurement
(of level means) for Walls of Widow's Creek No. 5

		Estimates Based Upon		
Furnace	Wall	Row Means	First Differences	Second Differences
A	Side	37.06	18.83	16.26
	Division	34.92	31.68	27.58
	Front	43.57	11.09	5.71
	Burner	68.56	46.97	28.80
B	Side	206.72	59.99	59.76
	Division	35.76	23.77	25.50
	Front	96.14	65.44	46.17
	Burner	36.24	46.17	58.03
(mean)		(69.9)	(36.7)	(33.5)
(median)		(40.3)	(39.3)	(28.2)

(A) always use 1st differences,

(B) always use 1st differences, except use 2nd differences for the burner wall of A furnace

(C) always use 2nd differences, except use 1st differences for the front wall of A furnace

(The value of 5.71 for 2nd differences and the front wall of A furnace seems quite sure to be artificially small. Its use would not be defensible.)

Rather less well defensible, though reasonably plausible, would be to combine estimates of variability across walls, using either of:

(D) The mean estimate based on 1st differences, 36.7, for all walls.

(E) The mean estimate based on 2nd differences, 33.5, for all walls.

Until we have a clearer understanding of the variability to be expected in the individual estimates, the writer would be a little reluctant to use such pooled estimates in highly controversial situations, even though he might regard one of their values as his best personal guess of the overall measurement variability involved in a typical level mean.

10. Equivalent Degrees Of Freedom

Before we can use our estimates of overall measurement variability effectively, we need a moderately good idea of how much variability is intrinsic in such estimates. The distribution of such estimates is not one of the most manageable of distributions, even under the most utopian assumptions. (It took the combination of World War II and the involvement of the great mathematician John von Neumann to start its mathematical investigation.) Accordingly it should be no surprise that our approach to the question of stability is today a little indirect.

The reason why we have to consider the "degrees of freedom" of a variability estimate in carrying out significance tests and finding confidence intervals is a simple one. If we have an estimate of variance that, like all real estimates, is subject to fluctuation, we must be prepared for the fact that the number we have before us may be either too large or too small. If we want to make statements with small probability of error, the consequences of our apparent variability being too small are very much more drastic than those of its being too large. Accordingly we have to increase somewhat the critical t-value (or the critical value of another appropriate statistic) in order to compensate for this.

For estimates based upon the sum of squares of deviations from a mean (or fitted trend) the well-founded practice is to count the number of linearly independent residuals, and use the resulting number of *degrees of freedom* in entering tables of critical values.

For the 5% point of Student's t, the effect of such care is fortunately not too large, the critical values being

2.57	for 5 degrees of freedom	(≈30%)
2.23	for 10 degrees of freedom	(≈15%)
1.96	for very many degrees of freedom	

It is, however, large enough that we dare not neglect it.

For estimates based on a conventional sum of squares of deviations, the relation

$$\text{degrees of freedom} = \frac{2(\text{Gaussian-average})^2}{\text{Gaussian-variance}}$$

holds exactly where "Gaussian-average" means the average, of our variability estimate, when the values that enter into it have a Gaussian distribution, and "Gaussian-variance" means the corresponding variance of our variability estimate. Accordingly, we do relatively well to define

$$\text{equivalent-}df \equiv \frac{2(\text{Gaussian-average})^2}{\text{Gaussian-variance}}$$

for other estimates of variability and to plan to use the resulting "degrees of freedom" in entering t-tables, etc.

The weakest link in this process is easily identified. We need to increase t because our estimate of variability may be *too small*. The greatest contribution to the Gaussian-variance of our estimates comes from situations where the estimate of variability is *too large*. However, to the best of today's insight, it appears likely that this loose linkage contributes more to excess conservatism than to excess liberality, especially in the case of mean square successive differences. So we tolerate it for the moment.

11. Equivalent-*df* For MSSD's

Since we can calculate equivalent-*df* for mean square successive differences from Table 1 of Morse and Grubbs (1947) in the form (expressed in their notation)

$$\text{eq-}df = (n-1)\cdot W(n;p)$$

with the results shown in Exhibit 3, and since we have today no better basis for incorporating information about the stability/instability of our variance estimates based on mean square successive difference, the recommended best practice, for the present, is to use values from Exhibit 3 in entering our t-table.

In the case of Exhibit 1, then, if we had chosen 1st differences we would have

estimated variance of a level mean: 59.99

estimated variance of grand mean: $5.45 = \frac{59.99}{11}$

estimated standard deviation of same: $2.33 = \sqrt{5.45}$

equivalent-*df* (from Exhibit 3): 6.90

corresponding t-value (by interpolation): 2.37

Exhibit 3

Values of eq-*df* for Mean Square Successive Differences
of Orders 1 to 4, Based on 3 to 15 Successive Values
(based on Morse and Grubbs, 1947)

# of Values	Order of Successive Differences			
	First	Second	Third	Fourth
3	1.60	1.00	xxxx	xxxx
4	2.25	1.38	1.00	xxxx
5	2.91	1.86	1.28	1.00
6	3.57	2.36	1.66	1.22
7	4.24	2.87	2.07	1.53
8	4.90	3.37	2.49	1.88
9	5.57	3.84	2.91	2.25
10	6.23	4.40	3.34	2.62
11	6.90	4.91	3.77	2.99
12	7.56	5.42	4.20	3.29
13	8.23	5.93	4.62	3.74
14	8.90	6.45	5.06	4.12
15	9.56	6.96	5.49	4.50

so that the ± associated with a 95% confidence interval, calculated as usual as

$$\pm(\text{critical value of } t)(\text{estimated standard deviation})$$

would be

$$\pm(2.37)(2.33) = \pm 5.54$$

We have thus completed an illustration of how we can use mean square successive differences to yield a confidence interval for a grand mean more appropriate than one based on *all* the differences among level means.

Note also that in our case we got 6.9 equivalent-*df* for 1st differences, but would get only 4.9 for 2nd differences. This is an example of the general reason for the maxim stated earlier, when in doubt use the lesser order of differencing.

C. Problems Of Multiplicity

12. The Root Of The Problem

In the situation arising in studying boiler corrosion, as in so many situations of science or engineering, we have to look at a number of more or less parallel results, where any one result may be trying to tell us about an important fact. Good engineering or scientific practice will direct our attention most heavily to the result that seems most important, something that would cause no additional difficulty if the precision of our result is very high.

The appropriate allowance to be made for uncertainty in a measurement depends on both an honestly-assessed measurement variance and the degree of selection for apparent importance involved. To know that Mr. X has a coin which has come up heads 10 times in a row is quite a different thing if:

(a) this is the only coin he ever flipped

(b) this is the 987th coin he flipped, each 10 times, and the only one to give 10 heads

(c) this is the 15,234th coin he flipped, each 10 times, and the only one to give 10 heads.

The chance of 10 heads, for a "fair" coin, independently flipped is one in 1024 ($=2^{10}$). Under circumstances (a) 10 heads is quite unlikely for a fair coin, and we have strong evidence that the coin is not fair. For 987 fair coins, all fair and independently flipped 10 times, the expected number of 10-head observations is $987/1024 \sim .96$, and one 10-head result is just what we might expect. In situation (b), then, there is no evidence against fairness. For 15,234 fair coins, all fair and independently flipped 10 times, the expected number of 10-head observations is $15,234/1024 = 14.9$; to have seen only one is a shocking deficit, and we have strong evidence that the successive flips, or successive coins, are not independent.

The data was the same, the selection situation was very different, and the answers were quite different.

13. The Confidence-Interval Case

In setting confidence intervals, in the usual situation where we have parallel results and must be interested in the apparently largest result, it is thus not satisfactory to merely set individual confidence intervals, each of which has, say, 5% chance of failing to cover the true value of what has been measured.

The simplest solution, which is not all that much more conservative than the other plausible solutions, is to set, in addition to *individual* confidence intervals, *simultaneous* confidence intervals so chosen that the average number of failures of any/all of the collection is .05. (Just as the average number of failures of any one individual 95% confidence interval is .05.)

This can be easily attained, when we have k parallel quantities, for each of which a confidence limit is desired, by looking up the critical value of the appropriate statistic, not for 5%, but for $5\%/k$. Once the average rate of failure of each confidence interval to cover what it should is set at $.05/k$, the average number of failures (of any confidence interval) is k times as large, namely .05. (No question of dependence or independence arises, because we are dealing with averages, and the average of a sum is the sum of the averages, willy-nilly.)

In the case of Widow's Creek No. 5, we have the ±'s for individual and simultaneous confidence limits shown in Exhibit 4. The change due to admitting that we must look at the largest value is large, but not so large as to be unreasonable.

D. Closing Comments

14. What We Have Faced Up To

All the points we have so far discussed are directed toward a single end:

- finding the most useful honest estimate of variability of our results.

Compared with the most naive — and incorrect — analysis of the Widow's Creek No. 5 data, one in which the same grand means are used for each of the walls (or half walls) but where

- all differences among results for the given wall are allowed to contribute equally to the assessment of overall variability
- the resulting measurement variance is divided by the total number of observations
- no allowance is made for the parallel analysis of several walls and half-walls.

We have alternately widened and narrowed our confidence intervals.

When we recognized the importance of calibration, we inevitably *widened* our confidence intervals. In particular, if, both before and after, 11 calibrations and 99 measurements are involved in the corresponding grand mean for some hypothetical wall the variance of the grand mean difference (= difference of grand means) is

$$\frac{\sigma_c^2}{11} + \frac{\sigma_m^2}{99} + \frac{\sigma_c^2}{11} + \frac{\sigma_m^2}{99} = .182\left(\sigma_c^2 + \frac{1}{9}\sigma_m^2\right)$$

so that if σ_c^2 is anywhere nearly as large as σ_m^2 the variance of the change will be dominated by the calibration variance — and we are stuck with the limited number of independent calibrations made.

When we recognized that there may be — in fact, almost inevitably are — trends from level to level in the time changes in thickness, and accordingly gave up s^2 (among level means) in favor of overall-measurement variance estimates based on mean square successive difference, we inevitably *narrowed* the confidence intervals for wall means. In Exhibit 2, the estimated variances were reduced by a factor of roughly 2, indeed once greater than three, thus narrowing the confidence limit to not quite about 70% — in one instance, not quite about 55% — of their former length. ("Not quite" because of loss of degrees of freedom.)

When we recognized the inevitability of looking at the most important of several results, and complemented our individual confidence intervals by simultaneous confidence intervals, we inevitably *widened* our confidence intervals.

Through all these changes of building in greater usefulness and honesty in our estimates of uncertainty — it is important to notice — the grand mean for each wall was left unchanged. This we took as given, and tried to do as well as we could in fitting it out with an appropriate measure of uncertainty.

15. Auxilliary Comment

In practice, especially on the burner wall, the number of measurements made in a level is not the same for all levels. We have abstained from recognizing this explicitly. Instead we have treated alike the level means for any given wall, making no explicit allowance for how many observations were made on each level. Beside simplicity and clarity of analysis, there are a number of reasons why this is reasonable, and perhaps even necessary. In particular:

(A) The variability of any one level mean is dominated by the variance of the associated calibration (since with roughly 4 to 10 measures per level we are dealing with something between $\sigma_c^2 + \frac{1}{4}\sigma_m^2$ and $\sigma_c^2 + \frac{1}{10}\sigma_m^2$ and since both reason and data analysis indicate $\sigma_c^2 > \sigma_m^2/2$). Thus the variances of the level means are much nearer to being equal than they are to be inversely proportional to the number of measurements.

(B) Differences of the variances by factors of less than two rarely have important repercussions of any sort.

(C) The relation

$$\frac{1}{n}\,\text{ave}(s^2) = \text{var}(\bar{y})$$

applies so long as the averages of the y's are the same, and the y's are uncorrelated, however alike or different the variances may be. (The similar relation for mean-square-successive differences will be only approximate, but its approximateness arises mainly because of the behavior of the first and last levels, which levels play distinctive roles — and which levels usually have plenty of measurements.)

(D) An analysis based on estimating σ_c^2 and σ_m^2 separately, and then using the ratio of these estimates to assign weights to the different levels would be (a) at best complex, (b) less resistant and more subject to perturbation by oddities, (c) less well understood.

In view of all these points, choosing the equally-weighted mean of level means seems the course of wisdom and conservatism.

We could here have replaced, and might often need elsewhere to replace, means by modern robust/resistant estimates of location. All the same considerations would apply.

16. Calibration Checking

When is a "calibration" really a calibration for statistical purposes? Is putting the gauge on the test block and saying "Ahh, it's still the same!" a calibration? This depends on what is written down, and on how what is written down is used.

A calibration, statistically speaking, occurs when something is done which can shift all the measurements that follow — or precede. If we spin away the adjusting knob, and use the test block to help us set it back where it should be, that is surely a calibration. If we put the gage on the test block, and say "it's reading 0.3 mils high," this is a calibration if and only if we subtract the +0.3 mils from all the measurements to which this "calibration" applies. If we merely check the gauge on the test block, and say "it checks closely enough" we dare not consider this an independent calibration. For *no one* would argue that following a setting-adjustment calibration, taking 8 such checks would reduce the variance of our calibration by division by a factor of 9.

To measure better, when calibrations are needed and calibration variance is an important contributor to overall measurement, we have to have *more* calibrations that are *independently* arrived-at calibrations.

Reference

Morse, A. P. and Grubbs, F. (1947). "The estimation of dispersion by differences," *Annals of Mathematical Statistics* **18**: 193-214.

Rather more details of the analysis of the Widow's Creek No. 5 data will appear in the appendix of a forthcoming E.P.A. report.

Exhibit 1

Row Means and Two Successive Differences
For Apparent Thickness Change: Side Wall of B Furnace,
Widow's Creek No. 5

Level	Row Mean	First Difference	Second Difference
1	2.86		
		5.71	
2	−2.86		18.57
		−12.86	
3	10.00		−18.57
		5.71	
4	4.29		3.57
		2.41	
5	2.14		11.43
		−9.29	
6	11.43		−36.43
		27.14	
7	−15.71		28.57
		−1.43	
8	−14.29		−7.85
		6.43	
9	-20.71		−2.14
		8.57	
10	−29.29		13.57
		−5.00	
11	−24.29		
Sum of Squares	2597.81	1199.89	3226.82
(sum)	(−76.43)		

Examples: 5.71 = 2.86−(-2.86), −12.86 = −2.86−(10.00), ...
18.75 = 5.71−(−12.86), − 18.57 = −12.86−(5.71), ...

Note: The choice of algebraic sign for the differences can be arbitrary, so long as it is consistent.

Exhibit 4

Illustration of Effect of Changing from Individual to Simultaneous Confidence Intervals, Apparent Changes in Tube Thickness, Widow's Creek No. 5

A) FULL-WALL MEANS

Furnace	Wall	± for Confidence Interval	
		Individual	Simultaneous
A	Side	±3.1	±5.1
	Division	±3.8	±6.1
	Front	±2.6	±4.3
	Burner	±5.2*	±8.8**
B	Side	±5.6	±9.0
	Division	±3.3	±5.3
	Front	±5.5	±8.8
	Burner	±4.3	±6.9

Note: Simultaneous intervals are for $k=8$.

* ±4.6 if *second* successive differences used.

** ±8.3 if *second* successive differences used.

B) A few HALF-WALL MEANS

Furnace	Wall	Half	± for Confidence Interval	
			Individual	Simultaneous
A	Side	Top	±4.6	±7.4
	Front	Top	±3.4	±5.7
		Bottom	±3.4	±5.7
B	Side	Bottom	±8.2	±13.0
		Top	±5.0	±7.5

Note: Simultaneous intervals are for $k=16$.

SESSION III:

SPECIAL TOPICS

DAVID G. LACHAPELLE
CHAIRMAN

PANEL:

COMBUSTION SOURCE/AIR POLLUTION REGULATIONS -- PRESENT AND PROJECTED

1 -- **Federal Regulations**
Jack R. Farmer

2 -- **Regional Regulations**
Robert Duprey

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FEDERAL REGULATIONS

By:

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ABSTRACT

The key provisions of the Clean Air Act which affect stationary sources of combustion are discussed. Primary emphasis is placed on section 111, standards of performance for new stationary sources. The current standards and plans for future standards are also discussed.

INTRODUCTION

It's a pleasure to have this opportunity to discuss EPA's regulatory program for combustion type stationary sources. EPA gives high priority to regulating emissions from these types of sources, because they are major sources of emissions of sulfur dioxide, particulate matter, and nitrogen oxides.

The Clean Air Act contains several regulatory provisions; however, sections 110 and 111 are the most applicable to combustion type stationary sources. Section 110 pertains to State Implementation Plans and section 111 pertains to new source standards of performance. I will limit my comments to new source standards of performance, and Mr. Duprey, the next speaker, will discuss the State Implementation Plans. I will discuss the recent amendments to section 111 of the Act, the current standards of performance, and the future standards of performance.

CLEAN AIR ACT AMENDMENTS OF 1977

On August 7, 1977, President Carter signed into law the Clean Air Act Amendments of 1977.[1] The amendments represent two and one-half years of effort by the Congress. Extensive amendments to the Act were last enacted in 1970. At that time, Congress established a broad program to achieve steady reduction of pollutants from both stationary and mobile sources. The 1977 Amendments are designed to continue that program, and, at the same time, provide a mechanism to continue economic growth.

The 1977 Amendments address many complex and controversial issues which have developed as a result of implementing the 1970 Amendments. I will briefly review the amendments pertaining to section 111, new source standards of performance.

In presenting the amendments to the Senate, Senator Muskie[2] said:

> "Under the Clean Air Act Amendments of 1971 [1970], section 111 requires that EPA promulgate performance standards reflecting the best system of emission limitation for new sources. Congress had several reasons for including this requirement. First, standards with a degree of uniformity are needed to avoid situations where some States may attract industries by relaxing standards relative to other States. Second, stringent standards enhance the potential for long term growth. Third, stringent standards may help achieve long-term cost savings by avoiding the need for more expensive retrofitting when pollution ceilings may be reduced in the future. Fourth, certain types of standards for coal burning sources can adversely affect the coal market by driving up the price of low-sulfur coal or effectively excluding certain coals from the reserve base because their untreated pollution potentials are high. Congress does not intend that new source performance standards contribute to these problems. Fifth, the standard-setting process should create incentives for improved technology."

The 1977 Amendments made the following important changes to section 111:

1. By August 7, 1978, EPA is required to list the categories of major stationary sources which have not already been listed and regulated under standards of performance. Regulations must be promulgated for these new categories on the following schedule:

25 percent of the listed categories by August 7, 1980
75 percent of the listed categories by August 7, 1981
100 percent of the listed categories by August 7, 1982

A Governor of a State may apply to the Administrator to add a category which is not on the list or to revise an existing standard of performance.

2. EPA is required to review the standards of performance every four years, and if appropriate, revise them. A separate provision requires EPA to promulgate revised standards of performance for fossil-fuel fired steam generators by August 7, 1978.

3. EPA is authorized to promulgate a design, equipment, work practice, or operational standard when an emission standard is not feasible.

4. The term "standard of performance" is redefined and a new term "technological system of continuous emission reduction" is defined. The new

definitions clarify that the control system must be continuous and may include a low-polluting or nonpolluting process or operation. They also require that standards for fossil-fuel fired stationary sources include both an allowable emission limit and a percentage reduction. Any cleaning of the fuel or reduction in the pollution characteristics of the fuel after extraction and prior to combustion may be credited in meeting the standards.

5. The Administrator is authorized to grant waivers from the requirements of standards of performance when innovative technology is used. To receive a waiver, an owner or operator must demonstrate to the Administrator that the proposed control system has not been adequately demonstrated; that its operation will not contribute to an unreasonable risk to public health, welfare, or safety; and that there is a substantial likelihood that the system will work effectively.

6. Regulations established by a State or EPA which cover designated pollutants (e.g. fluorides and sulfuric acid mist) must include consideration of the remaining useful lives of the sources.

7. Country grain elevators having a storage capacity of less than two and one-half million bushels are exempt from standards of performance.

8. Units that are ordered to convert to coal are exempt from the requirements of standards of performance.

9. States are authorized to enforce standards of performance against sources owned or operated by the U. S. Government.

We are reviewing our current program in light of these amendments and are planning the actions needed to implement the new requirements.

CURRENT STANDARDS

To date standards of performance have been promulgated for 24 source categories[3] and have been proposed for three additional source categories.[4,5,6] Emissions from five combustion type stationary sources are affected by these standards:

(1) Fossil-fuel fired steam generators.

(2) Municipal incinerators.

(3) Sewage sludge incinerators.

(4) Waste heat boilers and fuel gas combustion devices in petroleum refineries.

(5) Recovery furnaces in kraft pulp mills.

Standards of performance for fossil-fuel fired steam generators and municipal incinerators were promulgated in December, 1971. Several revisions have been made to the standards for fossil-fuel fired steam generators over the years. Most recently, a standard was proposed to limit nitrogen oxides emissions from the combustion of lignite. A final standard for lignite should be promulgated around November, 1977.

The standards for sewage sludge incinerators and combustion devices in petroleum refineries were promulgated in March, 1974. Standards for recovery furnaces in kraft pulp mills were proposed September, 1976, and should be promulgated around November, 1977.

In terms of legal actions against EPA, six of the 24 standards have been litigated. The six are (1) fossil-fuel fired steam generators, (2) sulfuric acid plants, (3) portland cement plants, (4) asphalt concrete plants, (5) primary copper smelters, and (6) primary aluminum reduction plants. Only two of these cases, portland cement plants and asphalt concrete plants, have been completely resolved.

In the case involving the fossil-fuel fired steam generator standard, the U. S. Court of Appeals for the District of Columbia found that the record generally supported the standards, but remanded the following three issues for further consideration:

(1) the need for provisions that unavoidable excesses of the standards during startup, shutdown or malfunction not be considered violations;

(2) the adverse environmental effects of setting a standard that requires coal-fired power plants to use a lime slurry scrubbing system; and

(3) the reasonableness of the opacity standards.

The startup, shutdown, and malfunction issue was resolved in October, 1973, when EPA promulgated regulations which, in effect, suspend the standards during startups, shutdowns, and malfunctions. EPA's final response to the issue concerning the adverse environmental effects of disposing of the sludge generated by the use of lime slurry scrubbing systems will be published in the very near future. Part of the opacity issue is similar to one raised in the suit on the portland cement plant standards, where the court upheld EPA's position on opacity. The other part of the opacity issue involves the specific standard for fossil-fuel fired steam generators. We will address this issue along with the scrubber issue and promulgate a final opacity standard for fossil-fuel fired steam generators.

EPA has a Compliance Data System which is used to keep a record of the sources subject to Agency and State regulations. According to the information in this system, there are 1742 sources covered by the 24 final standards of performance. About 78 percent of these are in compliance with the standards. The remaining 22 percent are in various stages of compliance or noncompliance. The number of combustion type stationary sources covered are as follows:

141 fossil-fuel fired steam generators.
12 municipal incinerators.
29 sewage sludge incinerators.
10 waste heat boilers or fuel gas combustion devices.

FUTURE ACTIONS

Although 27 source categories have been listed, there are many other major stationary sources which will be listed and covered by standards of performance. As indicated earlier, the Clean Air Act Amendments of 1977 require EPA to establish standards of performance for all major stationary sources by August 7, 1982, and to review all existing standards at four year intervals and revise them if appropriate.

For stationary source combustion categories, our immediate plans are to list and develop standards for gas turbines, stationary internal combustion engines, and industrial boilers.

Standards of performance for gas turbines will be proposed in the Federal Register within the next few days. Standards for stationary internal combustion engines are scheduled to be proposed in the summer of 1978. Standards for industrial boilers are in the preliminary stages of development and a definite date has not been established for their proposal.

One of our most important projects is revision of the fossil-fuel fired steam generator standards. In addition to being required by the Clean Air Act Amendments of 1977, the Sierra Club and the Navajo Tribe petitioned EPA on August 6, 1976, to tighten the standards to require 90 percent reduction in SO_2 emissions from new power plants. We have begun a study to complete the technological, economic, and other documentation needed to make appropriate revisions to the standards for SO_2, particulate matter, and nitrogen oxides. Several contractors and EPA personnel are working on this project, and we plan to propose the revisions by early 1978. The major areas of investigation are: (1) assessment of the SO_2 removal efficiencies of flue gas desulfurization systems (FGD); (2) collection of continuous SO_2 emission data at the inlets and outlets of one to three FGD systems installed at domestic power plants over periods ranging from one to three months; (3) investigation of methods of sludge disposal and fixation, and techniques to treat waste water or reuse it within the power plant; (4) determination of the effectiveness of particulate matter control systems; (5) determination of the costs and economic impact of alternative standards and control systems; (6) assessment of the secondary environmental impacts of alternative control systems on energy and solid waste, water, and air pollution; and (7) assessment of the impact of alternative control systems on coal marketing. The necessary information on nitrogen oxides control systems has already been collected.

Another important but complicated project under development involves defining the requirements of the current SO_2 standards for fossil-fuel fired steam generators when low sulfur coal rather than a FGD system is used. The standard does not specify the sampling and analysis procedures or the averaging times to use with low sulfur coal. We plan to propose regulations in the next few months which will establish requirements for these areas. This project addresses the current standard and is not directly involved in the project to revise the standards.

SUMMARY

In summary, new source standards of performance are an integral part of the air quality management approach established by the Clean Air Act. Basically, the air quality management approach is required for the attainment and maintenance of national ambient air quality standards. Although State Implementation Plans based on current knowledge must be pursued under the Act, mastery of air pollution caused by pollutants such as particulate matter and oxidants may be a longer term affair and may eventually depend simply on widespread application of good source control. In this regard, new source standards of performance operate like automobile standards. They are the least costly way of applying source controls over the long term. They allow consideration of economic impact at the time standards are set, they minimize the possibility of continually changing emission limitations, and they provide information on control needs at the time industry makes decisions.

REFERENCES

1. Conference Report on H. R. 6161. (H. R. ept. 95-564), "Congressional Record - House", August 3, 1977. pp H 8507 - H 8559.
2. "Congressional Record - Senate", August 4, 1977. pp S13701.
3. Chaput, L. S., Federal Standards of Performance for New Stationary Sources of Air Pollution, A Summary of Regulations, "APCA Journal, November 1976. pp 1055 - 1060.
4. Standards of Performance for New Stationary Sources, Kraft Pulp Mills. Federal Register, Vol. 41, No. 187, September 24, 1976. pp 42012 - 42028.
5. Standards of Performance for New Stationary Sources, Grain Elevator. Federal Register, Vol. 42, No. 9, January 13, 1977. pp 2842 - 2848.
6. Standards of Performance for New Stationary Sources, Lime Manufacturing Plants. Federal Register, Vol. 42, No. 85, May 3, 1977. pp 22506 - 22510.

EXISTING STATIONARY COMBUSTION SOURCE AIR POLLUTION REGULATIONS

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This is an opportune time to review the present and projected air pollution control regulations that affect combustion operations. On August 7, 1977, President James Carter signed into law the 1977 amendments to the Clean Air Act. These amendments will have far-reaching implications for combustion sources and fuel usage in this country. Some major shifts in policy as well as confirmation of present direction in policy is evident in the 1977 amendments. This paper reviews the past efforts at air pollution regulation of combustion operations and outlines how the amendments will affect stationary combustion sources.

1970 CLEAN AIR ACT AMENDMENTS

In order to explain the basis for U.S. Environmental Protection Agency (USEPA) policies affecting emission control requirements for stationary combustion sources, I would like to spend a few moments reviewing some of the provisions of the 1970 Clean Air Act as amended, because those amendments laid the foundation for cooperative Federal-state air pollution control programs as we know them today.

The Act directed the Administrator of the USEPA to establish ambient air quality standards for pollutants which were known to adversely affect human health and welfare. These standards were to reflect the maximum amount of specific pollutants which could be present in the air we breathe. National ambient air quality standards were set for sulfur dioxide (SO_2), total suspended particulates (TSP), carbon monoxide, (CO), photochemical oxidants (O_x), and nitrogen oxide (NO_x) in August of 1971. This paper is limited to regulatory programs aimed at controlling SO_2, TSP, and NO_x, since these are the pollutants of primary concern from stationary combustion operations.

The Clean Air Act went on to require each state to adopt, after public hearing, State Implementation Plans containing detailed strategies and emission control regulations designed to meet and maintain the National Ambient Air Quality Standards (NAAQS). Each plan had to be submitted to USEPA for review and approval. Most plans were approved by July of 1972. With respect to combustion operation, these state plans called for compliance with particulate matter and sulfur dioxide restrictions by July 1975, with provision for extensions to be granted up to July 1977. Except for the Los Angeles and Chicago areas, no NO_x regulations were required for existing sources, since all other areas were attaining the NO_x ambient standard.

CURRENT PARTICULATE AND SULFUR DIOXIDE REGULATIONS

For particulate matter, current emission regulations range from about 0.8 pounds per million BTU of heat input to 0.05 pounds per million BTU, depending on the state, specific geographic area, and capacity of the combustion operation. For large power plants, high performance electrostatic precipitators or wet scrubbers are required to meet these limits.

For sulfur dioxide, emission limits vary considerably more from state to state for existing facilities. As examples, current Illinois regulations allow emission of 1.8 lbs. of sulfur dioxide per million BTU actual heat input in the major metropolitan areas of Chicago, Peoria, and St. Louis. In other areas of the State, maximum emissions limits are generated by a formula based on plant characteristics. The average emission limit is 5.0 lbs. of sulfur dioxide per million BTU with a maximum of 6 lbs. Pennsylvania, second only to Ohio in total coal consumption, allows 0.6 lbs. sulfur dioxide per million BTU in Pittsburgh and Philadelphia, and 1.8 lbs. per million BTU in medium size cities. Other areas of the State are limited to 4.0 lbs. per million BTU. West Virginia allows 1.6 lbs. per million BTU in Charleston and 3.1 lbs. per million BTU statewide. Regulations newly developed by USEPA for the State of Ohio allow 1.0 to 1.9 lbs. per million BTU in the Cleveland area and 2.0 lbs. per million BTU in Youngstown. Other areas of the State have allowable emissions which range between 1.2 and 6.0 lbs. per million BTU. Effective next year, the State of Michigan will allow a maximum of 1.91 lbs. sulfur dioxide per million BTU statewide. The emission limit (New Source Performance Standard) for new coal-fired power plants constructed after December 1971 is 1.2 lbs. sulfur dioxide per million BTU nationwide. This is a Federal requirement which does not vary state by state.

COMPLIANCE STATUS

Compliance with these emission restrictions has been characterized by mixed success. Within the midwest region of USEPA, where over 30% of the nation's industrial capacity is located, compliance by industry with particulate and sulfur dioxide restrictions is 85%. However, in the State of Ohio, overall compliance is only 60%. Many utility and industrial boilers are currently in violation of both particulate and sulfur dioxide requirements in certain areas.

The situation with respect to improvement in air quality is also mixed. The trend for both particulate matter and sulfur oxides is definitely improved. The population exposed to levels above the national particulate standards has been reduced from 45% to under 30% from 1970 to 1974. About 50% of the total suspended particulate sites, however, still remain above the standards. For sulfur dioxide, the annual ambient standard has been nearly attained in most areas of the country. A significant exception area is the State of Ohio, where annual averages violations are experienced in several major cities. Isolated short-term (24-hr.) violations also still exist near some major sources in various states. Overall violations of sulfur dioxide standards have decreased by an average of 30% since 1970. NO_x ambient levels have generally been below the annual ambient standard of 100 micrograms per cubic meter, but are approaching the standard in several major cities.

THE 1977 AMENDMENTS TO THE CLEAN AIR ACT

The 1977 amendments to the Clean Air Act will have far-reaching effects on air pollution regulation of industry, including the utility and industrial combustion operations. Areas cleaner than the national standards are governed by a system of air quality increments designed to prevent significant deterioration of air quality. Non-attainment areas will be subject to provisions calling for tightening of current emission restrictions while permitting growth only if the new facility can offset its increased emissions by providing for controlling existing facilities (known as the USEPA Offset Policy).

PREVENTION OF SIGNIFICANT DETERIORATION

The amendments require plan revisions by the states to prevent significant deterioration. The amendments establish three land classifications related to specific allowable increases in total suspended particulates and sulfur dioxide over background concentrations.

Class I increments are set to protect pristine areas, Class II to allow some development, and Class III to permit more intensive growth.

The amendments prescribe the following specific increments:

(b)(1) For any class I area, the maximum allowable increase of sulfur dioxide and particulate matter over the baseline concentration of such pollutants shall not exceed the following amounts:

Pollutants	Maximum allowable increase (in micrograms per cubic meter)
Particulate matter:	
Annual geometric mean	5
Twenty-four-hour maximum	10
Sulfur dioxide:	
Annual arithmetic mean	2
Twenty-four-hour maximum	5
Three-hour maximum	25

(2) For any class II area, the maximum allowable increase of sulfur dioxide and particulate matter over the baseline concentration of such pollutants shall not exceed the following amounts:

Particulate matter:	
Annual geometric mean	19
Twenty-four-hour maximum	31
Sulfur dioxide:	
Annual arithmetic mean	20
Twenty-four-hour maximum	91
Three-hour maximum	512

(3) For each class III area, the maximum allowable increase in concentrations of sulfur dioxide and particulate matter over the baseline concentration of such pollutants shall not exceed the following amounts:

Particulate matter:	
Annual geometric mean	37
Twenty-four-hour maximum	75
Sulfur dioxide:	
Annual arithmetic mean	40
Twenty-four-hour maximum	182
Three-hour maximum	700

The law establishes as mandatory Class I areas all international parks, national wilderness areas and national memorial parks over 5,000 acres; national parks over 6,000 acres in existence upon enactment, and any Class I area established through USEPA redesignation procedures before enactment.

Fossil fuel-fired steam generating plants and boilers greater than 250 million BTU per hour heat input are subject to the Act's significant deterioration provisions.

NON-ATTAINMENT AREAS

With respect to areas exceeding particulate matter and sulfur dioxide national ambient standards, a state may approve new construction permits prior to July 1979 only if the source applies USEPA's offset policy. That is, the increased emissions from the new facility must be offset by further reduction in emissions from existing facilities below the levels currently specified in the approved State Implementation Plan.

No later than January 1979, each state must submit to the USEPA a revision to their state plan to provide for attainment of the national ambient standards for particulate matter and sulfur dioxide and nitrogen dioxide, if required, no later than December 31, 1982. After July 1979, a state must have in effect an approved plan revision from USEPA in order to issue new construction permits in non-attainment areas. Provisions for carbon monoxide and hydrocarbon sources are somewhat different but are not discussed here, since these contaminants fall outside the purview of this paper.

SPECIAL PROVISIONS AFFECTING FUEL COMBUSTION

The amendments settle a number of controversial issues affecting fuel-burning sources. The amendments limit credit allowed sources for stack height and require existing combustion sources to use continuous emission control rather than intermittant or supplemental controls as final control strategies. Congress, in effect, has determined that flue gas desulfurization is feasible and demonstrated technology to meet required emission reductions for sulfur dioxide.

A most novel provision in the amendments gives a Governor or the President the authority to require compliance with State Implementation Plan requirements by continuing to use local or regional available coal. The purpose of this amendment is to prevent regional economic disruption or unemployment. The effect again is to require the use of flue gas desulfurization or equivalent technology rather than allow importation of natural lower sulfur fuel as the means of compliance.

It should be clear that the 1977 amendments will provide new impetus and direction to the national air pollution control effort. Greater technological innovation will be necessary to meet these new requirements. This can be best stated by a quote from Senator Muskie in commenting on these amendments as follows:

> "This conference report is a composite that meets the public interest test. Most important of its provisions are, in my opinion, Senate requirements applicable to stationary sources. These provisions seek to put an end to the first round of efforts to circumvent emission control efforts by establishing new deadlines for existing industrial sources and penalties for failure to meet these new deadlines. There are clear messages in this Bill. The first message is to the Nation's

major industries. It can be taken from the amendment to which I just referred. That message is that the time for talk is over - the time for compliance is here. The health of the people can wait no longer."

SESSION IV:

STATIONARY ENGINE AND INDUSTRIAL PROCESS COMBUSTION SYSTEMS

JOHN H. WASSER
CHAIRMAN

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EMISSION CHARACTERISTICS
OF
SMALL STATIONARY DIESEL ENGINES

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INTRODUCTION

The Combustion Research Branch (CRB) of EPA's Industrial Environmental Research Laboratory-RTP has recently initiated both contract and in-house investigations for stationary reciprocating internal combustion engines. This paper will describe the initial phase of the in-house research program in this field. The work was directed at increasing understanding of air pollutant emissions from small stationary diesel engines and involved the use of emulsified fuel and a catalytic reactor as emission control methods to be evaluated.

EXPERIMENTAL APPROACH

A series of experiments were designed to evaluate the emissions from a small diesel engine, the effects of emulsified fuel on these emissions, and the emission characteristics of a catalytic reactor incorporated in the engine exhaust system. Table I outlines the experiments conducted. All experiments were run in random order and daily baselines were determined to eliminate atmospheric condition effects on emission levels.

EXPERIMENTAL SYSTEM COMPONENTS

The diesel engine used for this study is the prime element in an electrical generator package. This engine is a Caterpillar model D334, indirect injection, turbocharged and aftercooled, 4 stroke cycle, six cylinder diesel with 638 in.^{3} (1.045 X 10^{-2} m^3) displacement rated at 165 kW electrical output for continuous service. Other pertinent data includes:

Bore: 4.75 in. (120.6 mm)
Stroke: 6.00 in. (152.4 mm)

Compression Ratio: 17 to 1
Dry Weight: 2390 pounds (1084 kg)
Standard Timing: 14° BTDC

From the combustion standpoint, the important design characteristic is the precombustion chamber. The prechamber to total TDC volume ratio is approximately 0.3. In addition, the fuel injection pump output is controlled to maintain a constant engine speed (1800 RPM) under varying work loads. There are four in-head valves (two inlet - two exhaust) per cylinder. The engine is water cooled.

The catalytic reactor used in this study is an Engelhard Industries PTX-643D. This unit is a platinum metal catalyst deposited on a unitary porous ceramic support with a honeycomb construction. The catalyst is enclosed by an Inconel band and a stainless steel outer housing.

Ambient air is drawn into the machine's air inlet by the cylinder action and by the turbocharger. The turbocharger maintains an inlet manifold pressure of 16 psig (110.3 kPa) when operating at the rated full load, and is powered by the engine exhaust flow. Air is compressed by the piston action to approximately 250 psig (1723.5 kPa). Fuel is injected at approximately 650 psig (4481.1 kPa). Combustion gases flow from the cylinder chamber through the turbocharger and a muffler (either conventional or catalytic) to the exhaust stack.

Engine load is provided by an AVTRON model K463 load bank connected to the electric generator. This equipment provides electric load demands up to 225 kW in 5 kW increments.

The fuel used in the engine is a No. 2 diesel oil with an API gravity of 34.5 degrees. The bound nitrogen content of this fuel is only 0.04% (wt), thus fuel nitrogen conversion to NO_x is not a major factor in the emission levels measured in this study. The sulfur content is 0.12% (wt), sufficient to create the potential for significant sulfate

(SO_3, SO_4) emissions when a catalytic reactor is used with the engine to control CO and hydrocarbons. Other fuel properties were:

Carbon: 87.05% (wt)
Hydrogen: 12.71% (wt)
Ash: Trace
Higher Heating Value: 19,663 BTU/lb (45.74 MJ/kg)
Density: 0.8354 gm/cm^3 @ 100° F (311.1K)
Viscosity: 2.24 centipoise (2.24 mPa · s)
@ 100° F (311.1K)

ANALYTICAL SYSTEM

Methods for determining the gaseous and particulate emissions from the diesel engine were identical to those previously used by the author in gas turbine, boiler, and experimental combustor studies. Table II lists the instruments used for each flue gas component measured.

In addition to these methods, sulfates were determined using the Goksoyr-Ross controlled condensation coil. This method collects both SO_2 and sulfates. The total sulfur accountability with this method ranged between 80 and 85% based on the fuel sulfur content of 0.12% (wt). These sulfate measurements were made for CRB by IERL-RTP's Process Measurements Branch personnel.

The original, small-diameter, horizontal exhaust pipe on the engine was replaced by a larger-diameter stack extension downstream from the muffler. The extension was a 12 ft (3.66 m) high, 12 in. (304.8 mm) diameter vertical pipe. A sampling port in the extension was used for all particulate and sulfate traverse samples.

IMPORTANT OPERATING CHARACTERISTICS

SOUND LEVELS

Noise levels emanating from the diesel were measured to determine the sound exposure that would be experienced by personnel operating the engine and sampling equipment. The decibel level was found to change with electrical load on the machine. Two locations were selected for sound measurement: (1) 4 ft (1.22 m) from the diesel's weather enclosure with the access doors closed, and (2) at the surface of the enclosure with the access doors open. Table III lists the results of the survey made with a standard sound meter using the "A" weighting to relate to human sound perception.

These measurements indicate that, in the immediate area of the engine, the maximum permissible personnel noise exposure time per 24 hour day would only be 27 minutes without ear protection. In view of these levels, all personnel in the area are required to wear ear protection at all times when the engine is running.

PRECOMBUSTION CHAMBER

The precombustion chambers incorporated in this engine are a unique feature that could have a significant effect on emission performance. Each of the six cylinders has a small cylindrical chamber mounted above and connected to the main chamber by a nozzle-like passage. The fuel injection nozzle and glow plug for each cylinder are mounted with their functional sections inserted in the precombustion chamber. This prechamber design essentially performs the same function as staged combustion hardware in a boiler. All of the fuel for each firing cycle is injected into the precombustion chamber. Since only a fraction (3/10) of the combustion air is available in the prechamber, ignition and initial burning take place under fuel-rich conditions. As the rich burning mixture expands through the nozzle into the main chamber, the rest of the combustion air becomes available and combustion continues until quenched by gas expansion or wall effects. The primary effect of the prechamber system is a moderation of the peak pressures and temperatures generated in the engine, both of which influence emission levels.

LOAD PERFORMANCE

The diesel engine used in this study operates with large quantities of excess air. Excess air changes with load because both the air and fuel rates change. Atmospheric conditions (ambient temperature, pressure, and humidity) also affect the mass of air inducted by the engine, resulting in daily variations in excess air levels. Table IV contains a typical set of excess air values and exhaust temperatures recorded over the load range of the engine.

At full load (165 kW), a typical fuel rate was measured at 100.6 lb/hr (45.6 kg/hr). Using the net heat value of the No. 2 diesel fuel 18,400 BTU/lb (42.8 MJ/kg), the efficiency of the generator set was computed to be 38.8%.

EMISSION CHARACTERISTICS

BASELINE

Nitrogen Oxides

The initial investigation of the diesel engine consisted of measuring the emissions produced by the engine over its entire load range. These measurements were made before any operation or hardware modifications were attempted to establish the baseline characteristics. This baseline then served as a comparator for determining the performance of any potential emission control methods.

Baseline emission characteristics for oxides of nitrogen (NO_x) are presented in Figure 1. The actual data curve indicates that, as load is added to the machine, NO_x increases rapidly, experiencing a 104% rise as it reaches a peak at 40 kW. With further load increase, the NO_x emission rate decreases rapidly, reaching a minimum at 120 kW that is approximately 15% higher than the zero load level. As load is increased to the full load condition (165 kW), the NO_x emission rate increases slightly to a level that is 27% higher than the zero load condition. The full load NO_x level is 38% below the peak value measured at 40 kW.

This NO_x emission pattern presents a different situation than would normally be expected in that most combustion systems tend to produce NO_x maximums at or near full load. The increasing quantity of fuel being burned in the fixed volume of the combustion chamber continually increases the combustion intensity and temperatures involved, usually resulting in increased NO formation. However, in this engine, the two stage combustion mechanism changes the NO_x pattern.

To gain a better understanding of these results, a multiple linear regression curve fit was calculated using equivalence ratio (E.R.) and exhaust temperature as the independent variables. This two-variable regression line is shown on Figure 1 for comparison with the actual data points. The correlation coefficient (R^2) for this regression line was calculated to be 0.88, indicating a reasonable fit but certainly leaving room for improvement. The equation for this regression line is

$$[NO_x] = 63.0 - 0.0568\ [T_{ex}] - 4.54\ [E.R.]\ ,$$

where $[NO_x]$ is the NO_x concentration in grams/kg of fuel, $[T_{ex}]$ is the exhaust gas temperature in Kelvin, and E.R. is the equivalence ratio based on actual air divided by stoichiometric air requirements.

Another multiple linear regression curve fit was calculated adding the specific humidity (H) as an independent variable in the form of the natural antilog of the reciprocal of the humidity ($e^{1/H}$). This three-variable regression line is also shown on Figure 1 for comparison. The R^2 for this regression line was calculated at 0.967 indicating a significant improvement in fit. The equation for this regression line is

$$[NO_x] = 78.53 - 0.0983\ [T_{ex}] - 4.834\ [E.R.] + 2.43\ [e^{1/H}]\ ,$$

where H is in kg of water/kg of fuel.

By way of analysis, the use of these three variables has led to a reasonable NO_x prediction equation for this engine. However, the exhaust temperature and E.R. are only "stand-ins" for the real controlling factors. The exhaust temperature continually increases with load and thus is a reflection of the thermal regime in the cylinder combustion chamber. That is, the exhaust temperature has a mathematical function response to load comparable to the combustion zone thermal regime which is actually responsible for the NO_x formation. E.R. continually decreases with load and in that response is mathematically comparable to the effect of staged combustion created by the precombustion chamber design.

Referring again to the actual data curve, the rapid increase in NO_x from Zero to 40 kW can be considered the result of the thermal regime's being in control of NO_x formation. This is plausible in that the relatively small quantity of fuel injected is finding sufficient oxygen in the prechamber to burn fuel lean or near stoichiometric. As the fuel injected per firing cycle increases, there is no longer sufficient oxygen available in the prechamber and the staged combustion effect begins to reduce the NO_x formation.

The fraction of the total air charge per firing cycle required for stoichiometric combustion was calculated for each load setting. This data is listed in Table V. An engine prechamber was also measured: its volume is approximately 30% of the total volume of the combustion chamber at maximum compression. From this data, a stoichiometric quantity of air is available in a prechamber up to about 30 kW. This fact tends to support the postulated effect of staging previously discussed.

The effect of humidity is an inverse relationship with peak flame temperature, which in turn directly effects NO_x emission; thus, the use of $e^{1/H}$ gives a reasonably realistic mathematical response for this factor in NO_x formation.

Carbon Monoxide and Hydrocarbons

Baseline emission characteristics for CO are presented in Figure 2. The actual data points indicate that CO levels are a maximum at zero load and decrease rapidly as load is added to the machine, until 60 kW is reached. From that point the CO emissions are nearly constant to the full load point.

A regression curve was calculated using the natural log of E.R. as the independent variable. This regression line is shown on Figure 2 for comparison with the actual data points. The R^2 for this regression line is 0.99, indicating a close fit. The equation for the line is:

$$[CO] = 0.853\ [E.R.]^{1.74}.$$

The combustion conditions of higher temperatures and combustion intensity with increasing load contribute to better oxidation of CO. Wall quenching effects and relatively short residence times contribute to the continuing emission even at high load.

Baseline emission characteristics for gaseous hydrocarbons (defined as organic compounds that do not condense at 35° F (275K)) are presented in Figure 3. The actual data points follow a pattern similar to the CO data. Again, a ln vs. ln regression curve using E.R. as the independent variable resulted in a satisfactory curve fit. The R^2 was 0.946 for the equation

$$[HC] = 0.127\ [E.R.]^{1.133}\ .$$

Basically, the same conditions that control CO emissions are also responsible for the hydrocarbon emissions. The hydrocarbon emissions are significantly lower than the CO emissions and do not constitute a major emission problem for this engine.

Particulates

Particulate data was available for only full load operation of the diesel engine. The average filtered particulate loading was measured as 1.9235 gm/kg of fuel. In addition, condensable particulates amounted to 0.605 gm/kg in the benzene soluble fraction and 0.489 gm/kg in the water soluble fraction. The benzene soluble fraction is a rough estimator for potential emission of high molecular weight hydrocarbons, many of which are carcinogenic. Cascade impactor tests revealed that 65.35% of the particulate is less than 3μm, which is in the critical respirable range. Particulates remain a subject for further study with this diesel engine.

EMULSIFIED FUEL

Nitrogen Oxides

Because water addition to a combustion process is a known NO_x control technique, it was the first emission control method to be studied on this small diesel engine. Water and fuel were metered separately before mixing and emulsification in a gear pump. The emulsified fuel was injected into the engine through the standard fuel system;

no other modifications were needed. A surfactant was not used in these tests because emulsion stability was adequate without one.

The effect of water-fuel emulsification is presented in Figure 4. In general, the reduction in NO_x has a 1 to 1 relationship with the water to fuel (W/F) volume ratio. That is, an increase in W/F produces an equivalent decrease in the observed NO_x to base NO_x ratio (designated R). The amount of water that the engine would tolerate varied with load. The maximum W/F for a given load is given in Table VI.

At each load level, significant reductions in NO_x can be achieved by the water addition. At the peak NO_x emission load, 40 kW, the NO_x was reduced by almost 68%. At higher loads, 80 and 120 kW, the NO_x was reduced by over 80%. At full load, the machine would only tolerate sufficient water to reduce the NO_x level by about 45%.

In all likelihood, the NO_x reduction mechanism is the same for this engine as for other combustion systems. Temperatures are reduced, particularly the peak temperatures in the combustion zone, resulting in the reduction in NO formation.

Carbon Monoxide

Water injection with the fuel also affects the emission levels of CO. This data is given in Table VII.

There is an increase in CO at all load levels, although this increase is only 10% at full load. At other load levels, the CO increase ranges from double to 5 times the level without water addition. This increase is significant in the load range around 40 kW where the NO_x peak would most require water injection for control. Again, the reduced temperatures (which are beneficial in reducing NO_x) result in increased CO levels.

CATALYTIC CONVERTER

Effects of the catalytic reactor on engine emissions are presented in Figure 5. NO_x emissions were increased by 12 to 20% with a slight rising tendency with load. Emissions of both CO and gaseous hydrocarbons were drastically reduced. Hydrocarbons were reduced over 80% and CO over 90% from the baseline at each load condition except zero. At zero load the temperature averaged about 345° F (447K). This is below the effective operating temperature of the catalyst.

In addition to the increase in total NO_x, the NO_2 to NO_x ratio increased. This data is listed in Table VIII. With the exception of zero load, there was a substantial increase in the proportion of NO_2 in the total NO_x emission. Since the function of the catalytic reactor is to promote oxidation of the combustible flue gas components, it is not too surprising to find that NO and NO_2 were formed also.

Experience with automotive catalyst formation of sulfuric acid in exhaust effluents mandated a determination of sulfate emissions from the diesel because the diesel fuel contains substantially more sulfur than does gasoline. The sulfate tests were made only at full load operation. Baseline (without catalyst) sulfate level was near the detection limit of the procedure and was determined to be 2.5 mg/m^3 (0.025 gm/hp-hr). The sulfate level with the catalyst was measured at 123.1 mg/m^3 (1.23 gm/hp-hr), indicating a significant increase. Without the catalyst, only 1.1% of the total sulfur oxides was in sulfate form; the remainder was SO_2. With the catalyst, 59.15% of the sulfur oxides was in sulfate form. Again this is not unexpected because the oxidation catalyst should readily convert SO_2 to SO_3.

CONCLUSIONS

Based on these results with the Caterpillar D334 engine, the major air pollution problems for small diesel engines are the NO_x, CO, and fine particulate emissions. The precombustion chamber design was effective in controlling thermal NO_x emissions above the 40 kW load point.

Water-fuel emulsification had a mixed effect on the emissions: NO_x was dramatically reduced (60-80%), but CO emissions were increased by factors of 2 to 5. Further study of effects on particulates and fuel consumption is needed.

The catalytic converter reduced CO emissions by over 90% and hydrocarbons by over 80%. The averse effects of the converter included a 15 to 20% increase in NO_x and the creation of a significant sulfate emission that was not present in the unmodified engine.

TABLE I. EXPERIMENTAL PROGRAM

Experiment Set	Variables	Study Range
Baseline	Generator Load	zero to 165 kW at ∿ 20 kW intervals
Fuel Emulsion	Generator Load Water/Fuel Ratio	zero to 165 kW at ∿ 40 kW intervals zero to engine limit at 1 gph (3.785 Lph) water intervals
Catalytic Reactor	Generator Load	zero to 165 kW at ∿ 40 kW intervals

TABLE II. ANALYTICAL INSTRUMENTS

Flue Gas Constituent	Analyzer
O_2	Beckman Paramagnetic
CO_2	Beckman NDIR
CO	Beckman NDIR
Gaseous HC	Beckman Flame Ionization
NO/NO_x	TECO Chemiluminescent
Particulate Mass	RAC Staksamplr
Particulate Size	Andersen Cascade Impactor

TABLE III. NOISE LEVELS

Load, kW	dbA @ Location 1	dbA @ Location 2
zero	92	-
40	94	-
80	94	-
120	94	-
165	95	111

TABLE IV. EXCESS AIR AND EXHAUST TEMPERATURE WITH LOAD

Load, kW	Excess Air, %	Exhaust Temperature, °F (K)
zero	520.4	345 (447)
20	287.2	458 (510)
40	180.4	565 (569)
60	152.2	662 (623)
80	133.9	760 (678)
100	121.5	818 (710)
120	112.8	860 (733)
140	105.5	900 (755)
165	84.7	935 (775)

TABLE V. STOICHIOMETRIC AIR FRACTION WITH LOAD

Load, kW	Stoichiometric Air Fraction
zero	0.161
20	0.258
40	0.357
60	0.397
80	0.428
100	0.451
120	0.471
140	0.487
165	0.542

TABLE VI. MAXIMUM WATER/FUEL RATIO WITH LOAD

Load, kW	Maximum W/F
zero	0.459
40	0.589
80	0.875
120	0.642
165	0.353

TABLE VII. EMULSION EFFECT ON CO

Load, kW	Maximum W/F	CO/CO Base Ratio
zero	0.459	1.86
40	0.589	3.67
80	0.875	4.94
120	0.642	2.10
165	0.353	1.10

TABLE VIII. CATALYST EFFECT ON NO_2/NO_x RATIO

Load, kW	Baseline NO_2/NO_x	Catalyst NO_2/NO_x
zero	0.2105	0.1214
40	0.0361	0.1667
80	0.0244	0.3218
120	0.0180	0.2342
165	0.0471	0.1682

CONVERSION FACTOR

cubic inches X 1.6387 X 10^{-5} = m^3
inches X 25.4 = mm
pounds X 4.536 X 10^{-1} = kg
psi X 6.894 = kPa
BTU/lb X 2.326 X 10^{-3} = MJ/kg
gm/cm^3 X 1000 = kg/m^3
(^{o}F + 460)/1.8 = K
centipoise X 1.0 = mPa·S
gallons X 3.7854 = L
feet X 0.3048 = m

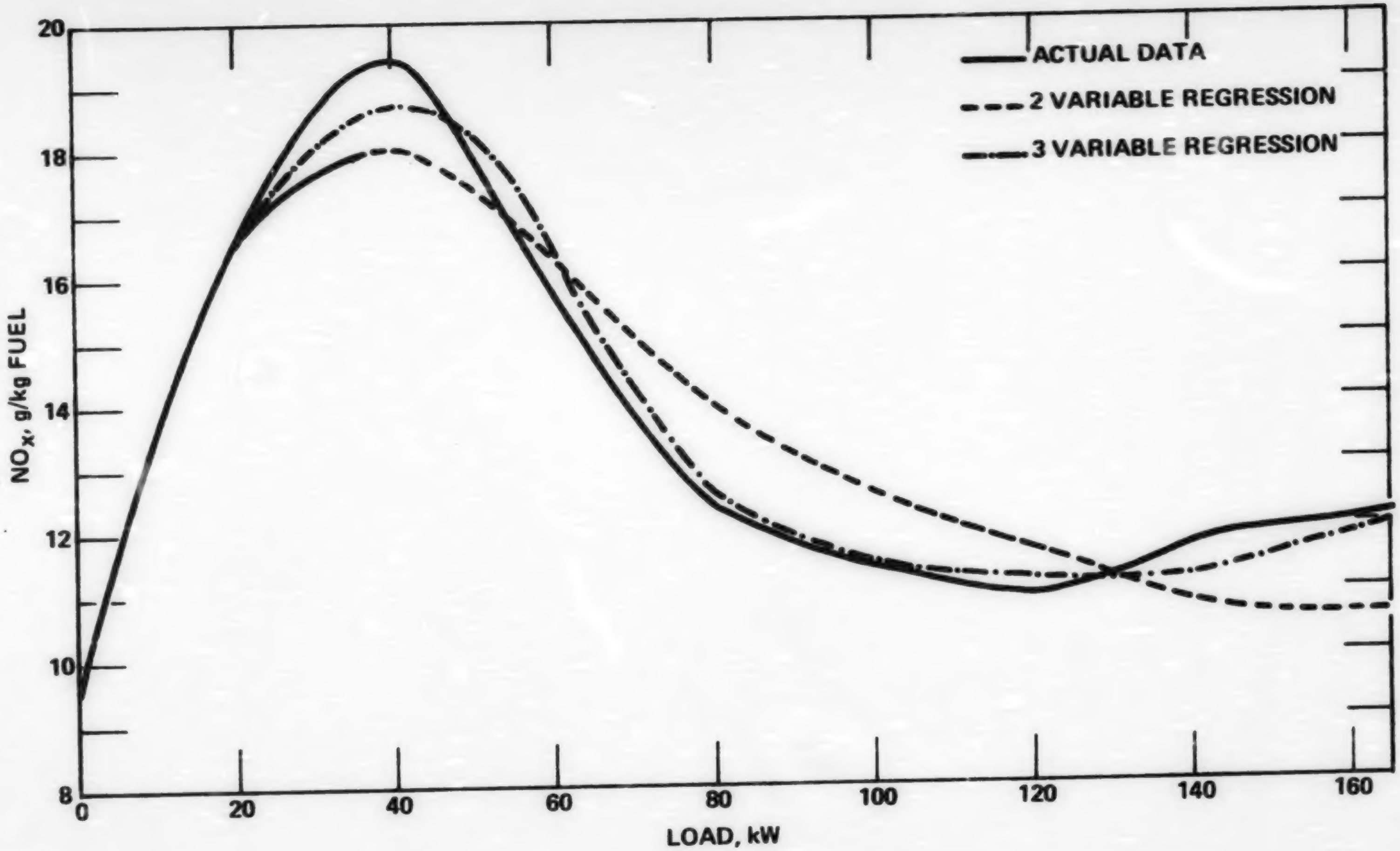

Figure 1. NO$_x$ emission characteristics of a precombustion chamber diesel engine.

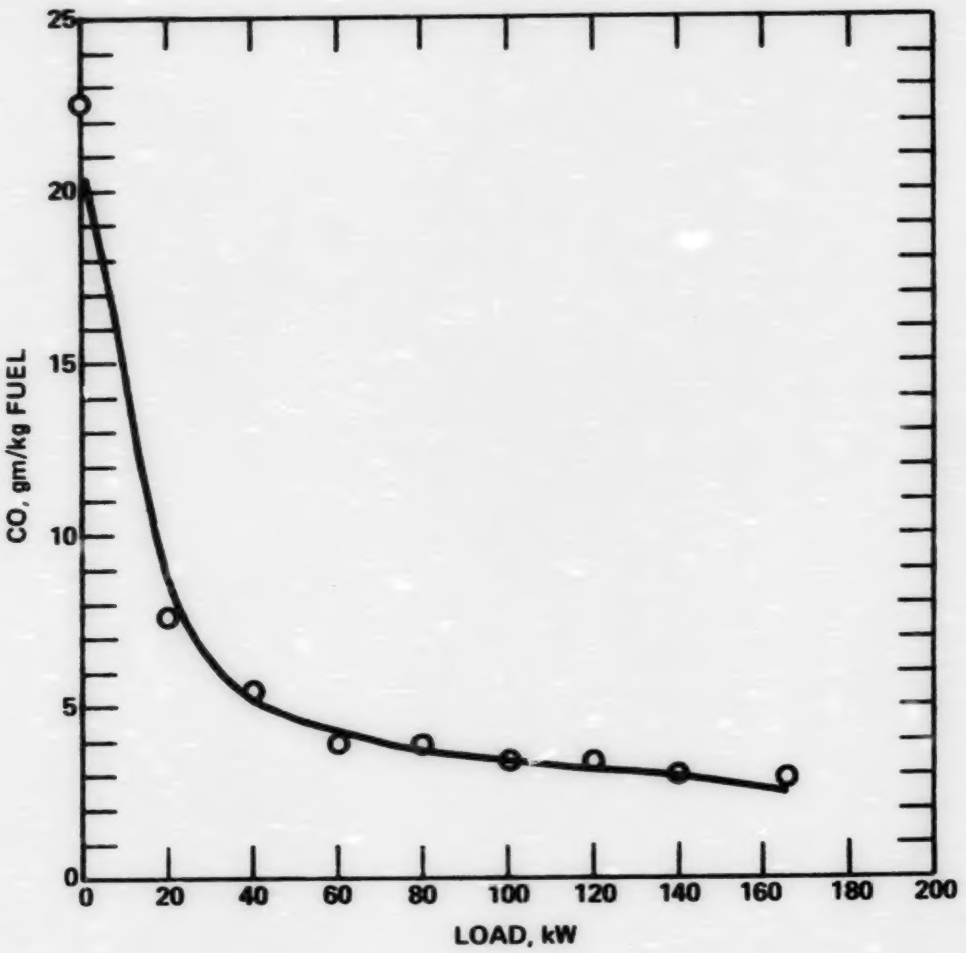

Figure 2. CO emission characteristics of a precombustion chamber diesel engine.

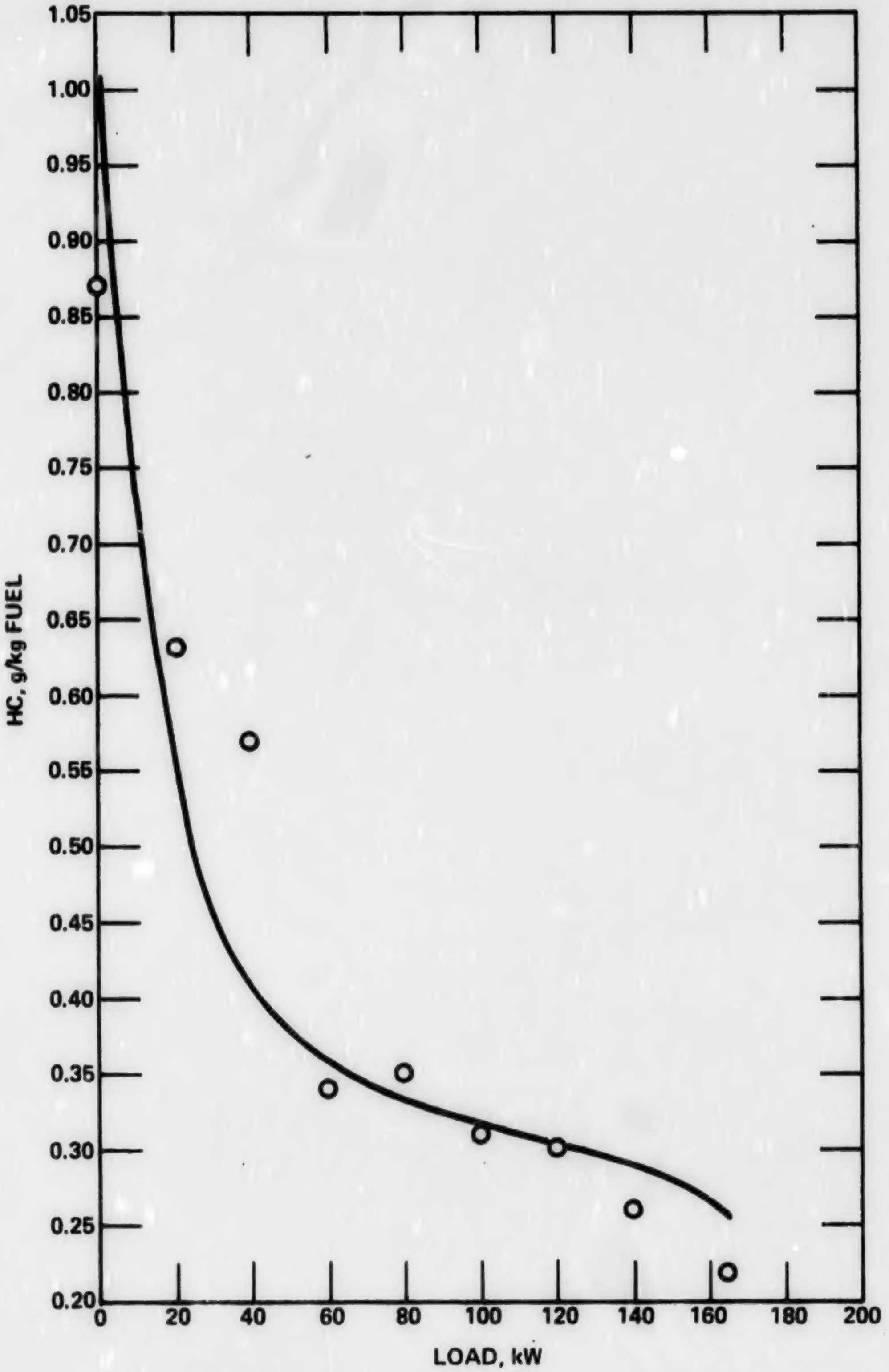

Figure 3. HC emission characteristics of a precombustion chamber diesel engine.

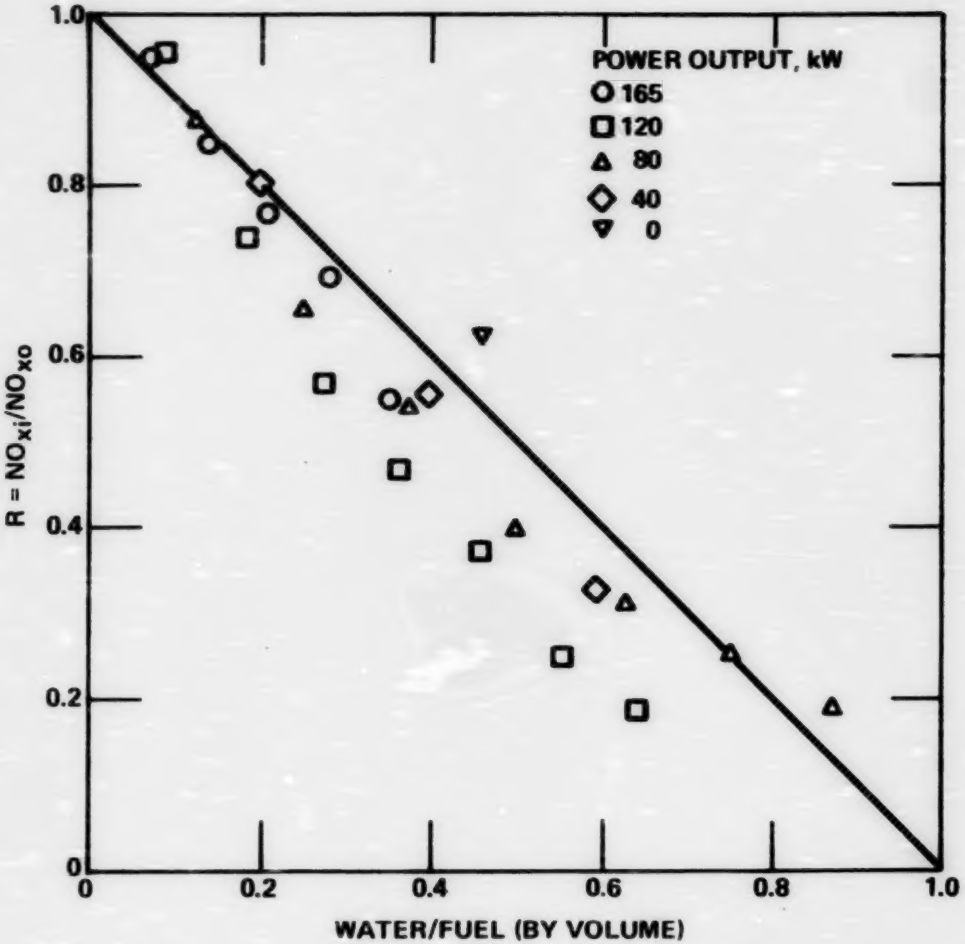

Figure 4. Effect of water-oil emulsion firing on NO_x emissions of a precombustion chamber diesel engine.

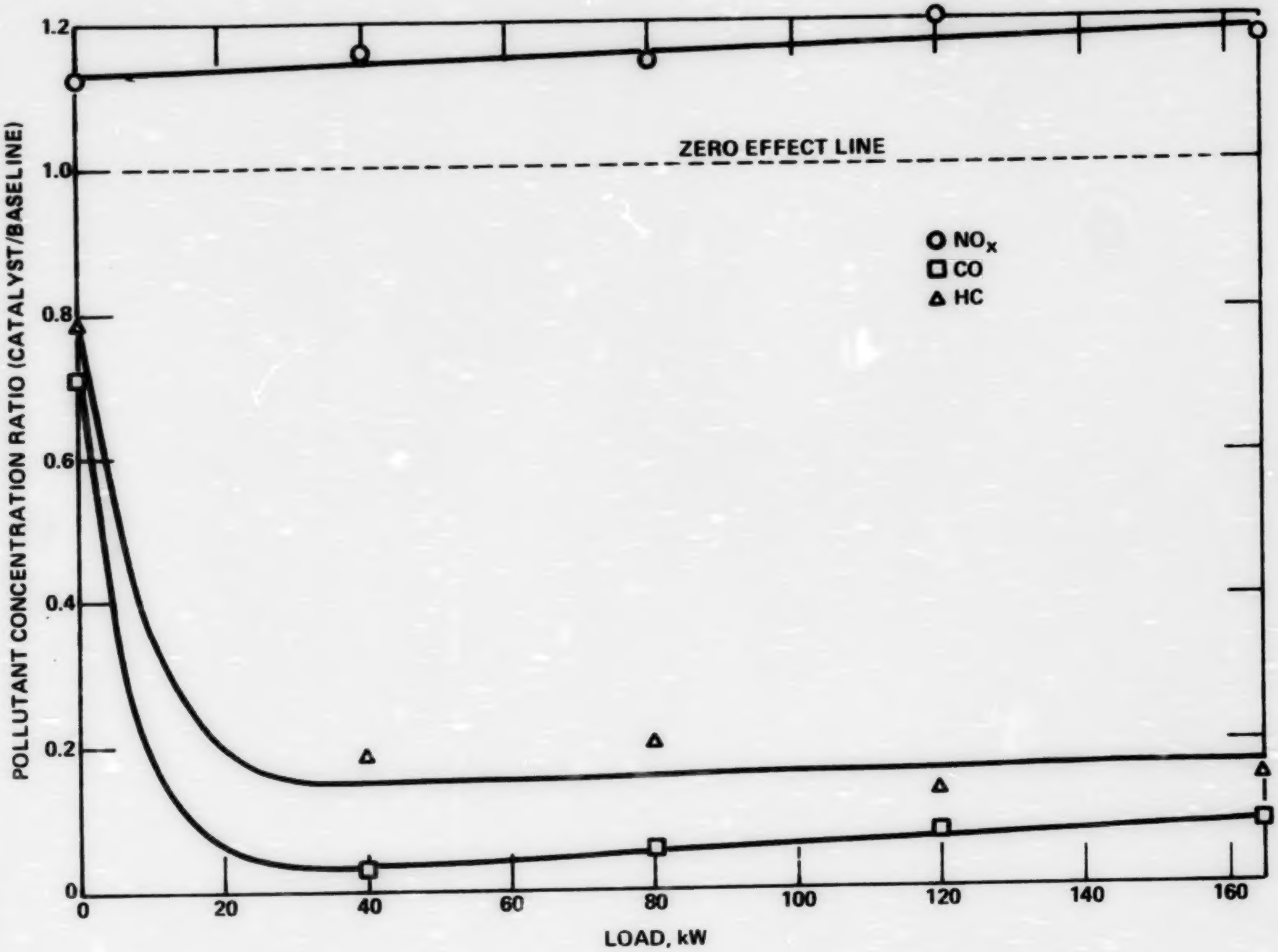

Figure 5. Effects of catalytic muffler on emissions from a precombustion chamber diesel engine.

SESSION V:

ADVANCED PROCESSES

G. BLAIR MARTIN
CHAIRMAN

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EVALUATION OF COMBUSTOR DESIGN CONCEPTS FOR ADVANCED ENERGY CONVERSION SYSTEMS

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SECTION 1

INTRODUCTION

This paper addresses the question of optimum combustor design for advanced power-generating systems incorporating an integrated coal gasifier. The utilization of coal via the production of a fuel gas is one alternative to conventional direct coal-fired steam boilers with stack gas cleanup. Environmental considerations and improved overall thermal efficiency are two of the attractive features of coal gasification when compared with conventional direct firing:

- Sulfur can be removed more readily from the fuel gas as hydrogen sulfide than can sulfur dioxide from the lean combustion products.
- Gasification of coal produces a fuel which can be utilized conveniently in a combined cycle giving an improved overall thermal efficiency even though there are energy losses associated with the gasification and fuel cleaning processes.

The improvement in plant performance offered by a gas turbine steam cycle power plant over a conventional steam cycle is associated with the ability of the combined cycle to utilize high temperature heat in the gas turbine and the exhaust heat rejected by the turbine to drive the steam cycle.

Figure 1 presents a simplified schematic of a basic coal gasifier, gas turbine topping, steam turbine bottoming (COGAS) cycle. Only the major heat, work and mass transfer paths have been shown for clarity. The single most important factor in the overall cycle efficiency is the turbine inlet temperature (TIT). Current maximum allowable TIT for industrial turbines is around $2200^{\circ}F$; however, improvements in design are expected to extend this limit to $2600^{\circ}F$ within a few years. The exhaust temperature from a typical state-of-the-art stationary gas turbine is approximately $1000^{\circ}F$. This exhaust gas enthalpy is used to generate steam in a waste heat boiler in the basic COGAS cycle. The fuel burned in the gas generator is produced by a pressurized gasifier and cleanup system. The properties of the fuel gas will depend strongly upon the gasifier design and cleanup process and less strongly upon the properties of the input coal. Neglecting the auxiliary steam raising plant required by the gasification process, the major gaseous emissions are nitrogen oxides and sulfur oxides. The level of the sulfur oxides are directly related to efficiency of hydrogen sulfide removal. Nitrogen oxide emissions are associated both with the product gas cleanup procedures and the gas generator (combustor) design.

Table I presents the results of a recent ECAS study to illustrate the projected NO_x emissions from COGAS systems. Estimates are also presented for a combined cycle burning a coal-derived liquid fuel. It should be noted that a fixed thermal efficiency of 74 percent was ascribed to the plant processing coal to manufacture a liquid fuel. The coal-derived liquid fuel was assumed to contain 1.3 percent by weight of nitrogen and both low Btu fuels contained ammonia. The fuel gas for Cycle 3 was produced in a multistage fluidized bed gasifier with in-bed desulfurization. It was estimated that approximately 15 percent of the nitrogen in the parent coal would appear in the product fuel gas in the form of ammonia. Cycle 2 utilized an advanced fixed bed gasifier and a cold gas cleanup system and produced a fuel which contained 0.04 percent by weight of ammonia. A similar plant designed to meet New Source Performance Standards (NSPS) for gaseous fuels would require a complex cleanup system to reduce considerably the H_2S and NH_3 content of the fuel gas. It should be remembered that the emissions listed in Table I are estimates and that three of them are lower than the NSPS for coal-fired steam plants. However, the NO_x emissions are considerably higher than advanced technical targets for power plants (3) Note that the Westinghouse efficiency estimates for Cycles 3 and 4 are considerably more optimistic than the General Electric estimates for Cycles 1 and 2. This is a consequence of different assumptions regarding component efficiencies and energy losses (gasifier, H_2S cleanup, compressor, turbine, blade cooling, cooling tower, etc.).

Two factors inherent in the design of high-efficiency COGAS systems tend to increase nitrogen oxide emissions.

- The use of high turbine inlet temperatures provide conditions conducive to thermal NO formation and techniques to lower flame temperature must be included in the combustor design.
- The use of high temperature sulfur cleanup processes does not remove the major portion of the ammonia from the fuel gas. Some fraction of this ammonia will be converted to NO during the combustion process.

The work discussed in this paper represents initial analytical studies which are the prelude to an experimental program designed to define optimum combustor designs for advanced power generation systems burning low Btu coal-derived fuel gases. The combustor designs are to be optimized to minimize pollutant emissions, but maximize overall plant efficiency. A series of limit-case studies have been carried out to establish the range of combustor conditions which are likely to minimize NO_x emissions and the effect on overall combined cycle efficiency of various combustor parameters has been assessed using a simplified cycle analysis program. The combustor parameters considered were those which would enable the emission of NO_x to be minimized either because NO formation was limited by the use of delayed mixing techniques to reduce fuel NO conversion or NO produced in a primary combustor was reduced in a secondary combustion system. Minimization of the impact of NO_x emission control on overall thermal efficiency or even enhanced thermal efficiency due to the application of control techniques were the main goals of the study. Inherent in this total approach is the belief that combustor design optimization offers an attractive NO_x control option to fuel denitrification for advanced power plants burning coal-derived fuels.

SECTION 2

LIMIT-CASE ANALYSIS TO DEFINE LOW NO_x COMBUSTOR DESIGN APPROACHES

It is well-known that the formation of NO by the oxidation of nitrogen compounds contained in the fuel can be restricted by causing the initial heat release to occur in fuel-rich regions. This is normally accomplished either by delaying the mixing of the fuel and air or by dividing the total air into two streams, one of which is injected downstram of the point of fuel injection. There are many practical examples of the use of these techniques to reduce emissions from existing coal, and oil-fired equipment. However, many of these examples utilized some ad hoc tuning method (biased firing, burners out of services, etc.) and do not necessarily represent the lower limit of NO production achievable with that particular combustion system. The limit-case studies being carried out as part of this project are planned to aid in the establishment of a lower limit of NO_x formation for a given set of combustor design constraints (e.g., size, pressure, heat extraction and firing low Btu fuel gas).

Limit-case situations are those in which some single aspect of the physics or the chemistry dominates and the remaining phenomena can either be ignored or modeled by simple idealized processes. The term is used in the context of this study to describe the methodology used to search for the lower bounds on NO_x emissions which are set by chemistry. Implicit in this approach is the assumption that all physical processes associated with fuel-air-product contacting and heat transfer can be accomplished in an ideal and optimum manner dictated by the chemistry. Having established this chemical-lower-bound the question then remains of how close one can approach the optimum physical transport behavior in real systems in order to achieve this lower bound.

Fundamental to the success of this approach is an authoritative description of the finite rate chemistry pertinent to the formation of nitric oxide, both from molecular nitrogen and nitrogen compounds such as ammonia contained in the fuel. Therefore, it is particularly important to describe both the generation and destruction of nitrogen species and their equilibration paths in rich mixtures. Nitrogen specie formation cannot be decoupled from the heat release mechanism and the kinetic mechanism used in the limit-case studies must adequately describe the mechanism of fuel oxidation and the reactions between hydrocarbon radicals and species containing nitrogen. Since low Btu fuel gases contain sulfur species allowance should be made for fuel NO_x/SO_x interaction.

The amount of ammonia contained in coal-derived low Btu fuel gases will depend upon the gasifier design, the coal-to-steam ratio, and upon the fuel gas cleanup system (1, 2), and this ammonia might well be the major source of NO. The parameters controlling the conversion of NH_3 to NO in combustion systems are:

- reaction zone stoichiometry,
- flame type, and
- fuel type.

Experiments with premixed flat flames tend to indicate that fuel NO conversion is relatively insensitive to flame temperature (4). Comparison of various experiments suggests that for fuel/air mixtures close to stoichiometric fuel nitrogen conversion is greatest with premixed flames and least in laminar diffusion flames. Fuel nitrogen conversion in turbulent diffusion flames and well-stirred reactors tend to fall between these two extremes illustrating the importance of the fuel/air contacting process to fuel NO formation.

The calculations discussed later in this section were carried out with a mechanism that had been compared with data on the production of thermal and fuel NO conversion in a jet-stirred reactor (5). However, it cannot be claimed that this mechanism is a complete description of nitric oxide formation and destruction in combustion systems since all the specie concentrations necessary to validate the mechanism were not available. Before discussing some of the initial limit-case analyses it is informative to describe, in qualitative terms, the sequence of events which are likely to occur during combustion of fuels containing ammonia, particularly under fuel-rich conditions. Summarizing the experimental observations it appears that:

- nitrogen compounds (even NO) added to hydrocarbon flames are converted to HCN, and the fraction converted is probably dependent upon temperature; and
- rich combustion products contain HCN, NH_i and NO.

A plausible, but not necessarily correct, mechanism includes the initial breakdown of NH_3 via

$$NH_x + H \rightarrow NH_{x-1} + H_2 \quad (x = 1, 2, 3) \tag{1}$$

allowing NO production by such reactions as

$$N + OH \rightarrow NO + H \tag{2}$$

and CN compounds can be produced by reactions involving hydrocarbon fragments and molecular nitrogen or NO:

$$NO + CH \rightarrow HCN + O \tag{3}$$

$$N_2 + CH \rightarrow HCN + N \tag{4}$$

thus, accounting for the presence of HCN. The oxidation of HCN is most likely to occur in post-flame gases via the intermediate compound NCO

$$HCN + H \rightarrow CN + H_2 \tag{5}$$

$$CN + OH \rightarrow NCO + H \tag{6}$$

$$NCO + H \rightarrow NH + CO \tag{7}$$

thus, allowing the synthesis of ammonia. The final level of NO and N_2 appears to be dependent upon competition between two reactions, one forming NO by oxidation of a nitrogen specie and the other forming nitrogen from the same specie. Two possible candidates for this competitive path are:

$$N + OH \rightarrow NO + H \tag{8}$$

or

$$N + NO \rightarrow N_2 + O \tag{9}$$

The design of the minimum NO emission staged combustor will attempt to minimize the total concentration of nitrogen specie (NO, NCO, HCN, CN, NH_i)

in the rich combustion products. The intent would be to drive the system to equilibrium which would give a low nitrogen specie concentration. Equilibration times might be excessive unless the reactant temperature is high. An alternative approach might well be to use parallel staged systems involving the mixing of two rich streams under optimum conditions. Another alternative could involve the production of NO in one combustor which then supplies vitiated air containing NO to a secondary combustor. If, as is suggested by reactions (8) and (9), NO itself is a necessary precursor to N_2 production partial reburning might well reduce nitrogen specie concentrations to a low level, but perhaps not to their minimum value, relatively quickly. The presence of NO during the initial NH_3 breakdown might well enhance N_2 production in the initial reaction zone at the expense of HCN production under optimum temperature conditions.

An illustration of limit-case analysis is provided by a series of calculations carried out to define the range of conditions most likely to minimize NO production in two types of combustors burning low Btu gas. The combustors were:

- a 10 atmosphere gas generator feeding combustion products at 2800°F to a gas turbine, the high temperature low Btu fuel was assumed to contain 4000 ppm of ammonia; and
- a supercharged steam generator operating at 10 atmospheres providing exhaust products to a low temperature (2000°F) gas turbine. The fuel for this boiler was assumed to contain 500 ppm NH_3.

The reactor analogue used in the limit-case calculations for the adiabatic gas generator is presented in Figure 2. The primary stage consists of a short, well-stirred ignition reactor followed by a long residence time, rich hold-up zone to allow time to minimize the concentration of nitrogen species at the exit. The primary zone is followed by a secondary burnout zone to give the desired turbine inlet temperature. Temperature control in the primary zone is achieved by heat transfer to the tertiary air stream which is fed into the second burnout zone as dilution air.

One aspect of the limit-case calculations carried out with the reactor analogue shown in Figure 2 was to define those conditions which would minimize the bound nitrogen specie concentration at the exit of the rich primary hold-up zone. It was found that this minimum value was achieved with equivalence ratios between 1.3 and 1.45. Also, the nitrogen specie concentration was independent of the initial bound nitrogen content of the fuel. Consider the results presented in Figure 3 which show the effective NH_3 conversion for adiabatic primary reactors with equivalence ratios varying from 0.45 to 2.0. The effective NH_3 conversion is defined as the difference between the sum of the nitrogen specie concentration at a given reactor residence time for a fuel with and without NH_3 normalized by the NH_3 content of the initial doped mixture. When this ratio is zero the exit conditions of the primary adiabatic reactor are independent of the initial NH_3 content of the fuel. These conditions occur for residence time t, ≥ 200 ms and $0.8 \leq \phi \leq 1.7$. Thus an optimum rich combustor would operate at an equivalence ratio of approximately 1.35 and have a residence time greater than 200 ms.

Having minimized nitrogen specie concentration at the exit of the primary hold-up section, the major design challenge involves the addition of second stage and dilution air without the production of thermal NO. Figure 4 presents the results of a series of calculations investigating the influence of various secondary air mixing schemes on NO production. The curves drawn with long dashes represent the temperature and NO levels associated with instantaneous mixing, the NO rises rapidly in the first few milliseconds as the nitrogen specie are converted to NO as the mixture becomes fuel lean. After the initial rise NO increases slowly due to thermal NO production. The solid line represents secondary air addition over a 50 ms time period. A temperature excursion to 2300 K occurs and the NO concentration exceeds 1000 ppm before decreasing back to 850 ppm by final dilution, precisely those conditions which should be avoided. If the rich products were to be added to the air over a 50 ms period the dot-dash curve is the calculated result. Since fuel is added to air the mixture temperature increases slowly until the auto-ignition temperature is reached at approximately 40 ms. High temperatures do not occur and the final NO is the same as the instantaneously mixed case. The short dashed curves in Figure 4 simulate NO production in a turbulent diffusion flame with the second stage heat release distributed over a range of equivalence ratios.

The limit-case analogue used for a series of staged supercharge boiler calculations is presented in Figure 5. Heat transfer to raise steam provides an additional degree of freedom over the adiabatic gas generator since it allows simultaneous heat release and heat loss in the secondary zone, thus minimizing residence times at higher temperatures. Figure 6 illustrates the calculated NO concentration for three limit conditions. Curve 'a' represents a system with a 1 ms stirred reactor for ignition followed by a plug flow section with a heat loss distribution as shown. The fuel and air are premixed. Curves 'b' and 'c' show the effect of operating under staged combustion conditions. The calculated NO concentration shown by curve 'b' was obtained with a short residence time stirred reactor feeding a plug flow section with the secondary air addition distributed over the first 500 ms or the burnout reactor. In curve 'c' the rich products are held at an equivalence ratio of 1.15 in a plug flow reactor for 500 ms before secondary air is added. The increase in NO level associated with the addition of second stage air is due to NO formation from the bound nitrogen species that exit the rich reactor. This conversion occurs at the 800 ms station where the system becomes lean. The difference in the final NO concentration shown for curves 'b' and 'c' is due to the reduction in bound nitrogen specie associated with the increased residence time in the rich zone of system 'c'.

SECTION 3

COMBUSTOR PARAMETERS AND COMBINED CYCLE PERFORMANCE

In addition to the basic COGAS cycle shown in Figure 1, the performance of three other variants has been assessed; these are:

- COGAS Cycle with Duct Heater. Heat is added to the exhaust of the gas turbine increasing its temperature and allowing the performance of the steam cycle to improve. From an NO_x emission control viewpoint it provides an opportunity to pass NO formed in the gas generator through a reaction zone causing some reduction in the exhaust NO level. This reaction zone may be that associated with a diffusion flame or it may be a staged system encompassing a rich primary zone followed by dilution. The potential level of reduction is dependent upon the relative firing rate of the duct heater to the main combustor.
- COGAS Cycle with Nonadiabatic Gas Generator. Steam is raised in the primary gas generator to improve the efficiency of the steam bottoming cycle. Simultaneous heat extraction, through steam raising and secondary air staging can suppress peak secondary zone diffusion flame temperatures thereby reducing thermal NO generation. In addition, if it should become apparent that the optimum primary zone temperature is below adiabatic, then steam raising in this zone could result in lower fuel nitrogen conversion.
- COGAS Cycle with Reheat Combustor. Figure 7 shows a schematic of a reheat combustor COGAS cycle. Reheating increases the power produced by the gas turbine while raising the turbine exhaust temperature to improve cycle performance. This system offers several possibilities for NO_x control.

- The primary combustor may be operated with minimum control and the NO reduced in the reheat combustor much as in the duct heater above.
- Both the primary and reheat combustors can be designed as staged systems, but as they operate under different conditions their design will undoubtedly be different.
- The fuel used by the reheat combustor (produced by a separate mid-pressure gasifier) could be tailored for low NO production by maximizing the hydrocarbon content and minimizing the bound nitrogen content.

A generalized combustor concept was used for the assessment of the cycle performance. Each combustor is assumed to have its own separate gasifier and associated cleanup system operating at combustor pressure, thus allowing the relationship between all inlet and outlet streams to be calculated without specifying the design details of the gasifier. Coal with composition CH_xO_y, steam and air flow into the gasifier which operates at an equivalence ratio $\phi_1 > 1.0$ where ϕ is the mass ratio of coal to air divided by the same ratio at stoichiometric conditions. The dirty fuel gas flows through a cleanup system where heat is lost and contaminants are removed. The clean fuel gas is burned in the combustor at $\phi_2 < 1.0$ and some heat may be transfered to raise steam at this point. All generalized gasifier/combustor flowrates except coal were considered to be mixtures of ideal gases with constant specific heats.

The primary heat exchanger for each cycle was the waste heat recovery boiler whose minimum pinch point temperature differential was 50°F, the gas side pressure drop was 0.1 atm and the steam side pressure drop was neglected. Adiabatic counterflow heat exchange units were assumed to be used for the supercharged boiler, gasifier intercooler and recouperator with a minimum hot/cold side temperature differential of 50°F. The performance of these heat exchangers was determined by specifying the quantity of heat transfered, calculating the outlet temperatures and effectiveness.

A simple steam cycle with no reheat or interstage bleed feedwater heating was used in the analysis of the various COGAS cycles. The steam cycle was optimized to produce maximum efficiency for a given gas turbine exhaust temperature. Steam for gasifier injection is bled from the steam cycle at maximum temperature and throttled to gasifier pressure before injection. Feedwater makeup was supplied at condenser temperature and blowdown was neglected. This steam cycle is not necessarily optimum for this application; however, since the purpose of this analysis was to specify combustor operating parameters and not to predict precise overall cycle efficiency, the details of the steam cycle are of second order importance.

The influence of the following parameters on the performance of the basic COGAS cycle were assessed initially:

- Turbine inlet temperature (TIT)
- Pressure ratio
- Gas turbine component efficiencies
- Combustor pressure drop
- Coal properties
- Gasifier steam injection
- Gasifier intercooling (low temperature versus high temperature cleanup)

The first three parameters primarily determine the gas turbine operating point and their variation demonstrates the tradeoffs in gas turbine and steam turbine performance. The last four parameters are related to gasifier and combustor design. The analysis was carried out by independently varying specific parameters and calculating cycle performance while maintaining all other parameters constant. Table II lists the values used for the various cycle parameters unless the influence of that parameter was being determined and was varied as part of the analysis.

Figure 8 shows the effect of TIT and gas turbine pressure ratio on the performance of the basic COGAS cycle. Two extremes were chosen for TIT; 2000°F, typical of current state-of-the-art stationary gas turbines, and 2600°F which is projected for advanced systems. Since increasing TIT increases both gas turbine cycle efficiency and turbine exhaust temperature (which increases steam cycle efficiency) there is an increase in overall cycle performance. Figure 8 also shows that in the combined cycle there is not a strong effect of pressure ratio on overall cycle performance. Although the gas turbine efficiency is strongly affected by pressure ratio, the tradeoff between gas turbine and steam turbine efficiencies tends to flatten the performance curve for the combined cycle.

Table III shows the influence of component efficiencies on combined cycle performance. A base case pressure ratio of 24 was selected, and as expected decreasing component efficiencies from 85 to 75 percent decreased the gas turbine cycle efficiency and increased the turbine exhaust temperature. The increase in steam cycle efficiency and relative power level was sufficient to substantially counter the decrease in gas turbine efficiency. Thus, the combined cycle is less sensitive to component efficiency than a conventional stand-alone gas turbine.

Combustor pressure drop will affect the performance of the gas turbine cycle and the analysis carried out as part of this study indicated that a 10 percent pressure loss caused a 2.5 percent decrease in gas turbine efficiency. However, the pressure drop across the turbine also decreases and this increases turbine exhaust temperature and improves steam cycle efficiency. The net effect is a much smaller drop in overall combined cycle efficiency. Thus the corresponding decrease in overall cycle efficiency to the 10 percent pressure loss is only 1.2 percent. This decreased sensitivity to pressure drop gives considerably more flexibility in combustor design and any incremental increase in pressure drop caused by a low NO_x design will have a minimum influence upon overall cycle efficiency.

In the previous calculations the sulfur products in the gasifier offgas were assumed to be removed in a high temperature cleanup process with heat loss equivalent to 10 percent of the heating value of the parent coal. If a low temperature cleanup process were to be used, the sensible heat in the low Btu gas must either be lost to the environment or transferred to other portions of the combined cycle. Since gasifier offgas temperatures are usually well in excess of $1200^{o}F$, it is wasteful to dissipate the intercooling heat in the environment. A more economical alternative is to generate steam from the hot LBG and either inject the steam into the gasifier or operate a steam turbine power cycle. Figure 9 shows the effects of intercooling at 2000°TIT and pressure ratio 12.0. The best alternative would be to use the intercooling heat to operate a steam cycle. Such a cycle could operate at 36.9 percent efficiency, and since this is less than the combined cycle efficiency without intercooling, overall efficiency decreases with intercooling. A decrease in overall efficiency of 1.6 percentage points will result if the sensible heat in the gas is 20 percent of the total coal heat input. If the intercooling heat is used to raise steam for gasifier injection, overall efficiency drops 3.8 percentage points, and if intercooling heat is wasted it drops 8.9 percentage points.

The effects of intercooling at 2600°F TIT and pressure ratio 12.0 are shown in Figure 10. Using intercooling heat to operate a steam turbine cycle is again the best alternative, but the minimum losses due to intercooling are increased to 2.3 percentage points at sensible heat/total heat equal to 20 percent. Intercooling and low temperature cleanup may introduce other losses including additional process steam, auxiliary power requirements and the removal of water vapor through condensation.

The previous discussion demonstrated that basic COGAS cycle optimization requires degrading gas turbine cycle efficiency to improve steam turbine cycle efficiency. The turbine exhaust temperature from an optimized gas turbine cycle is too low to raise 1000°F steam. However, since the gas turbine cycle efficiency is only slightly reduced as pressure ratio varies from the optimum value, pressure ratio may be reduced increasing turbine exhaust temperature, steam cycle efficiency and overall combined cycle efficiency.

Heating the gas turbine exhaust in a direct-fired duct heater is an alternative to reducing pressure ratio. This allows the gas turbine cycle to operate at its optimum pressure ratio and the steam cycle to operate with high efficiency. The disadvantage is that fuel burned in a duct heater bypasses the gas turbine cycle.

Figure 11 shows the estimated performance of two COGAS cycles with duct heaters and high and low turbine inlet temperatures. The efficiencies are plotted as functions of duct heater firing rate normalized to total heat input. As the fraction of heat used in the duct heater increases from zero there is an initial decrease in the overall efficiency associated with bypassing the gas turbine. This is followed by an increase in overall efficiency associated with the rapid rise in steam cycle efficiency. The results for a 2600°F TIT are similar to the 2000°F TIT except that less heat must be released in the duct heater to reach otpimum conditions for overall cycle efficiency.

A nonadiabatic combustor supplying hot gas to the turbine and raising steam for the steam cycle provides an alternative to the use of a duct heater. The performance of a supercharged boiler 2000°F gas turbine combination is compared in Figure 12 with a basic COGAS as a function of pressure ratio. At a pressure ratio of 6.0 the turbine exhaust temperature without supercharging is sufficient to operate a 1000°F steam cycle and this is the optimum nonsupercharged pressure ratio. As pressure ratio is increased turbine exhaust temperature drops and supercharging is required to maintain steam temperature. Overall efficiency rises to a maximum of 41.42 percent at a pressure ratio of 10. This is 1.32 percentage points more than that achieved with a duct heater. The equivalence ratio at optimum is 0.328 which is very lean compared to previous low excess air supercharged boiler concepts. Furthermore, the heat transferred through the supercharged boiler to the steam cycle is only 7.7 percent of the total heat input to the cycle. Thus the optimum supercharged boiler could be designed much like an adiabatic gas turbine combustor with additional coils to raise steam.

Figure 13 is similar to Figure 12 except that the TIT has been increased to 2600°F. The optimum efficiency is achieved with almost zero supercharging and is only 0.1 percentage points higher than without supercharging. Thus the desirability of supercharging decreases at high TIT.

A reheat combustor functions much like a duct heater except it is moved upstream between turbine stages. Figure 7 presents a schematic diagram of a COGAS cycle with reheat. The reheat combustor has its own gasifier and cleanup system supplied with air bled off the appropriate compressor stage. The two major variables associated with a reheat combustor are the reheat pressure and temperature. For a given reheat pressure, increasing the reheat temperature increases the mean temperature of heat addition to the gas turbine (and hence its efficiency). However, reheat temperature is limited by the same material constraints that limit TIT.

In this analysis a COGAS cycle with 2000°F TIT, a reheat temperature equal to TIT and a reheat combustor pressure drop of five percent was investigated. Figure 14 shows the effects of reheat pressure with pressure ratio held constant at 16.0. The performance of a nonreheat cycle is also shown and is the limit as reheat pressure approaches main combustor pressure. However, since the pressure drop of the reheat combustor is zero in the limit, the extrapolation of the reheat curves gives sightly different performance than the nonreheat case. As reheat pressure is decreased from main combustor pressure, the turbine exhaust temperature rises improving steam cycle efficiency. Gas turbine cycle efficiency also rises because the increase in mean temperature of heat addition to the gas turbine cycle overshadows the increase in turbine exhaust temperature. As reheat pressure is decreased further, the relative importance of these two effects interchange and gas turbine cycle efficiency reaches a maximum at a reheat pressure of 7.6 atm and then declines. Steam cycle efficiency continues to increase with turbine exhaust temperature reaching a maximum (at the same point as overall efficiency) at a reheat pressure of 6.85 atm.

Figure 15 compares the performance of basic COGAS and COGAS with reheat cycles as functions of pressure ratio. The reheat pressure selected was the lowest which produced a 1000°F steam cycle (6.85 atm). Reheat improves both steam cycle and gas turbine cycle efficiency and the maximum overall efficiency is 44.60 percent at a pressure ratio of 20. This represents an improvement of 4.4 percentage points over the non-reheat cycle.

SECTION 4

CONCLUSION

This paper has been concerned with the initial tasks in a project designed to assess the possibilities of combustor design to control emissions from advance power plants burning low Btu gas. Gasification allows coal to be used in an environmentally acceptable manner and when the gasifier is integrated with a combined cycle power plant it also represents an extremely thermally efficient conversion system. It is known that NO_x formation can be limited by combustion modification techniques. This paper has reviewed the types of control techniques that might be used and also assessed the influence of combustor design parameters on cycle performance.

A simplified cycle analysis program has been used to assess the tolerance of COGAS cycle performance to variations in combustor design. The types of combustor considered in the analysis were:

- an adiabatic gas generator,
- a reheat combustor,
- a nonadiabatic gas generator raising steam, and
- a duct heater for supplementary firing the waste heat boiler.

The study indicates that COGAS cycle performance is not strongly dependent upon such design parameters as:

- combustor pressure drop,
- combustor pressure,
- combustor stoichiometry (corresponds to the amount of steam raised in the supercharged boiler, the degree of reheat, or the fraction of total heat released in the duct heater).

Indeed, cycle performance can be enhanced by the use of a reheat combustor or a nonadiabatic gas generator which raises steam. The combined cycle performance is tolarent of these combustor parameters because heat lost from the gas turbine is not completely lost to the overall cycle since it can be used to raise steam in the bottoming cycle which is one of the inherent advantages of the COGAS cycle.

Low Btu coal gas will probably contain ammonia, some of which will be converted to NO during combustion. The tolerance of the overall cycle performance to variation in combustor design parameters allows several combustion modification techniques to be explored to limit NO_x emissions from advanced COGAS power plants to minimal levels. These techniques include:

- staged main gas generator with temperature control via steam generation;
- staged reheat combustor for minimum fuel nitrogen conversion and incineration of NO formed in the main combustor; and
- staged or diffusion flame duct burners for minimization of fuel nitrogen combustion and incineration of NO formed in the main combustor.

REFERENCES

1. Corman, J.C. and Fox, G.R., "Energy Conversion Alternatives Study", General Electric Phase II Final Report, NASA-CR134949, Vol. I, December 1976.

2. Beecher, O.T. et al, Energy Conversion Alternative Study, Westinghouse Phase II Final Report, NASA 134942.

3. Martin, G.B. and Bown, J.S., Development of Combustion Modification Technology for Stationary Source NO_x Control Proceedings of National Conference on Health, Environmental Effects and Control Technology of Energy Use, EPA Report 600/7-76-002.

4. De Soete, G.G., "La Formation Des Oxydes, D' Azote dans la Zone D' Oxidation des Flammes D' Hydrocarbures, IFP, June 1975.

5. Heap, M.P., et al, paper presented at the 16th Symposium (International) on Combustion, Boston, August 1976.

TABLE I. SUMMARY OF NO_X EMISSIONS AND OVERALL CYCLE EFFICIENCY FOR OPEN CYCLE GAS TURBINE SYSTEMS (AFTER ECAS (1,2))

Cycle Description	NO_X Emissions g/MJ	(lb/M Btu)	Overall Cycle Efficiency %
1. Water-cooled gas turbine TIT 1922K using coal-derived liquid fuel (G.E.)	0.73	(1.7)	37.5
2. Air-cooled gas turbine TIT 1589K integrated low Btu gasifier (G.E.)	0.15	(0.35)	40.0
3. Air-cooled gas turbine TIT 1644K integrated gasifier (Westinghouse)	0.18	(0.65)	46.8
4. Ceramic gas turbine components TIT 1644K coal-derived liquid fuel (Westinghouse)	0.26	(0.6)	38.5

TABLE II. CYCLE ANALYSIS CONSTANTS

Gas Turbine (Topping Cycle)

Inlet temperature/pressure - 59°F/1.0 atm

Coal = $CH_{0.8}O_{0.1}$, 14,704.5 Btu/lb (dry, ash-free)

Constant gas specific heats

Integrated gasifier

Gasifier: mass steam/mass coal = 1.0

Gasifier steam supplied from steam cycle

Gasifier $\phi = 2.0$

Gasifier and cleanup system heat loss - 10% coal H.H.V.

Generalized combustor pressure loss = 5%

Compressor and turbine efficiencies = 85%

Waste Heat Recovery Boiler

Exhaust side inlet pressure - 1.1 atm

Pinch point temperature difference = 50°F

Stack temperature = 300°F

Steam Turbine (Bottoming Cycle)

No feedwater heating

No reheating

Condensor temperature/pressure - 101°F/1.0 psia

Maximum steam temperature = 1000°F

Maximum turbine moisture = 15%

Turbine efficiency = 85%

TABLE III. VARIATION OF SYSTEM BEHAVIOR WITH CHANGE IN COMPONENT EFFICIENCIES FOR BASIC COGAS CYCLE (TIT = 2600°F, PRESSURE RATIO = 24)

Parameter	Base Case	Lower Component Efficiencies	Units
Turbine and Compressor Efficiencies	85.0	75.0	%
Steam Cycle Efficiency	33.79	36.89	%
Gas Turbine Cycle Efficiency	35.37	21.58	%
Overall Efficiency	47.00	39.26	%
Equivalence Ratio	0.371	0.342	
Turbine Exhaust Temperature (°F)	1122	1296	°F

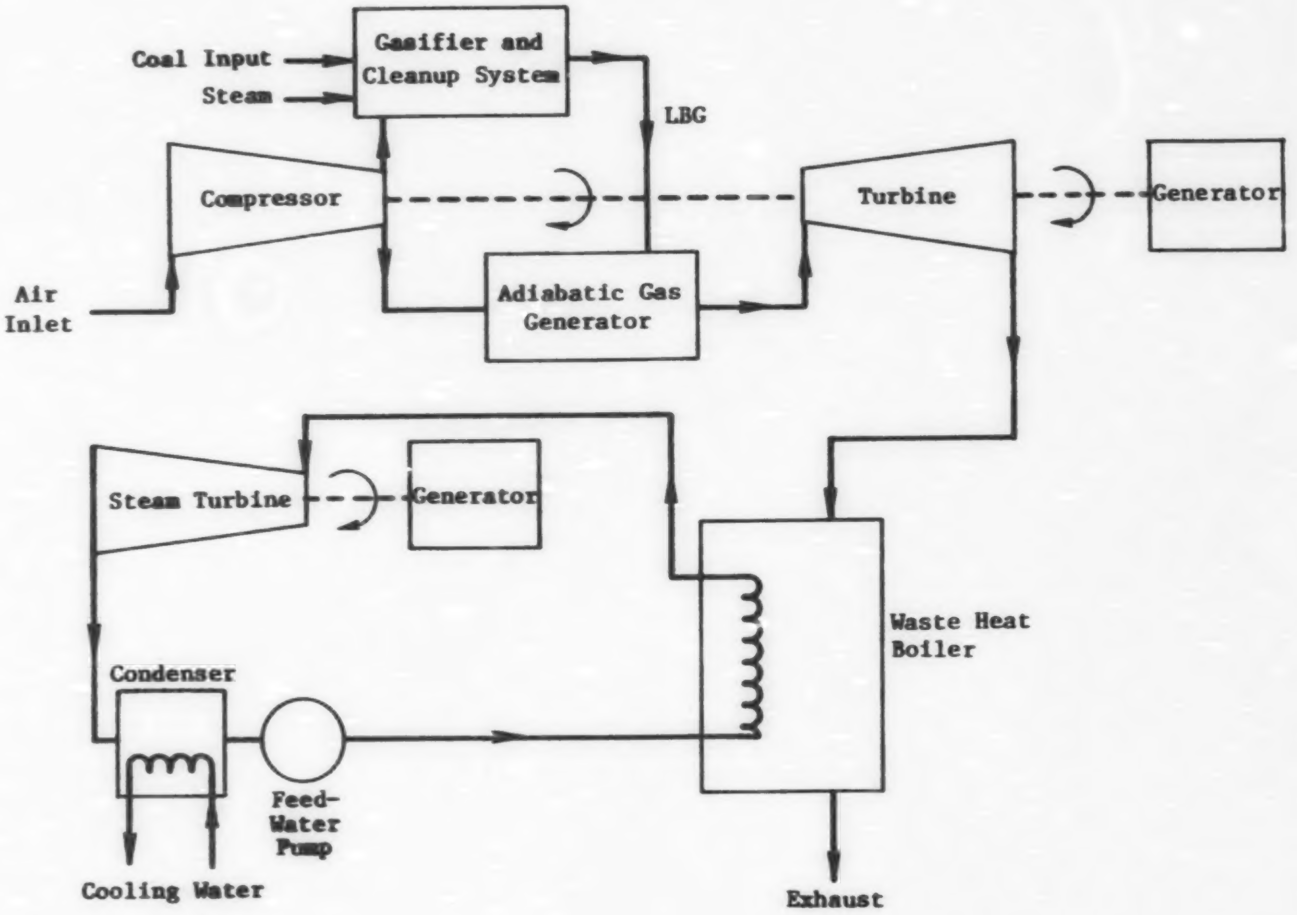
Coal Input
Steam
Gasifier and Cleanup System
LBG
Compressor
Turbine
Generator
Air Inlet
Adiabatic Gas Generator
Steam Turbine
Generator
Waste Heat Boiler
Condenser
Feed-Water Pump
Cooling Water
Exhaust

Figure 1. Basic COGAS cycle with integrated LBG gasifier.

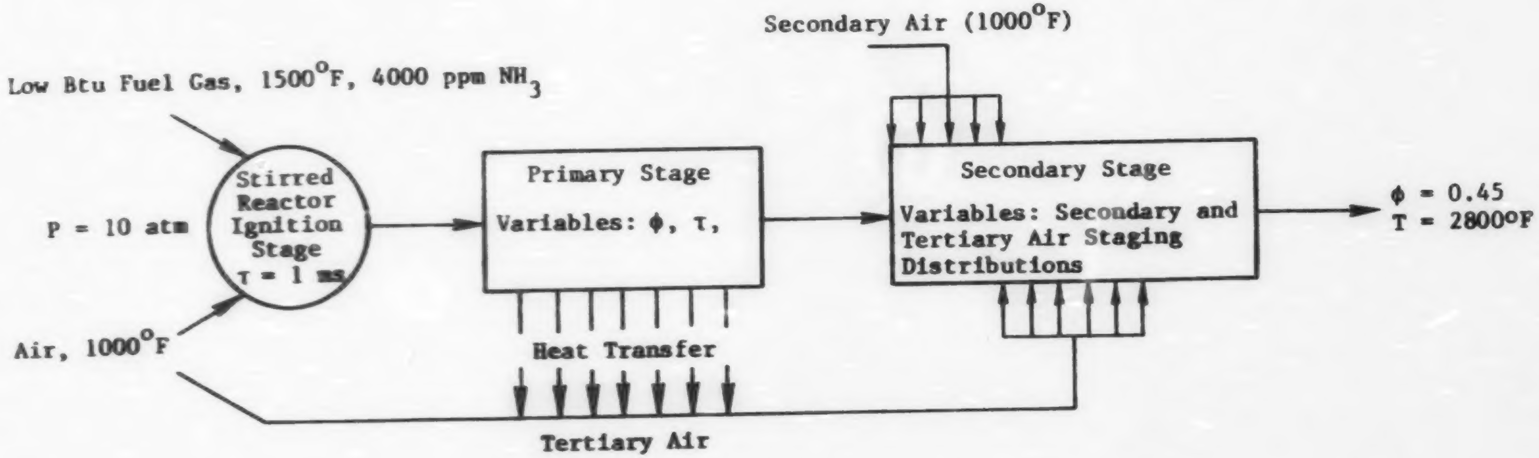
Low Btu Fuel Gas, 1500°F, 4000 ppm NH$_3$
P = 10 atm
Air, 1000°F
Stirred Reactor Ignition Stage
τ = 1 ms
Primary Stage
Variables: ϕ, τ,
Heat Transfer
Tertiary Air
Secondary Air (1000°F)
Secondary Stage
Variables: Secondary and Tertiary Air Staging Distributions
ϕ = 0.45
T = 2800°F

Figure 2. Limit-case analogue for the adiabatic gas generator.

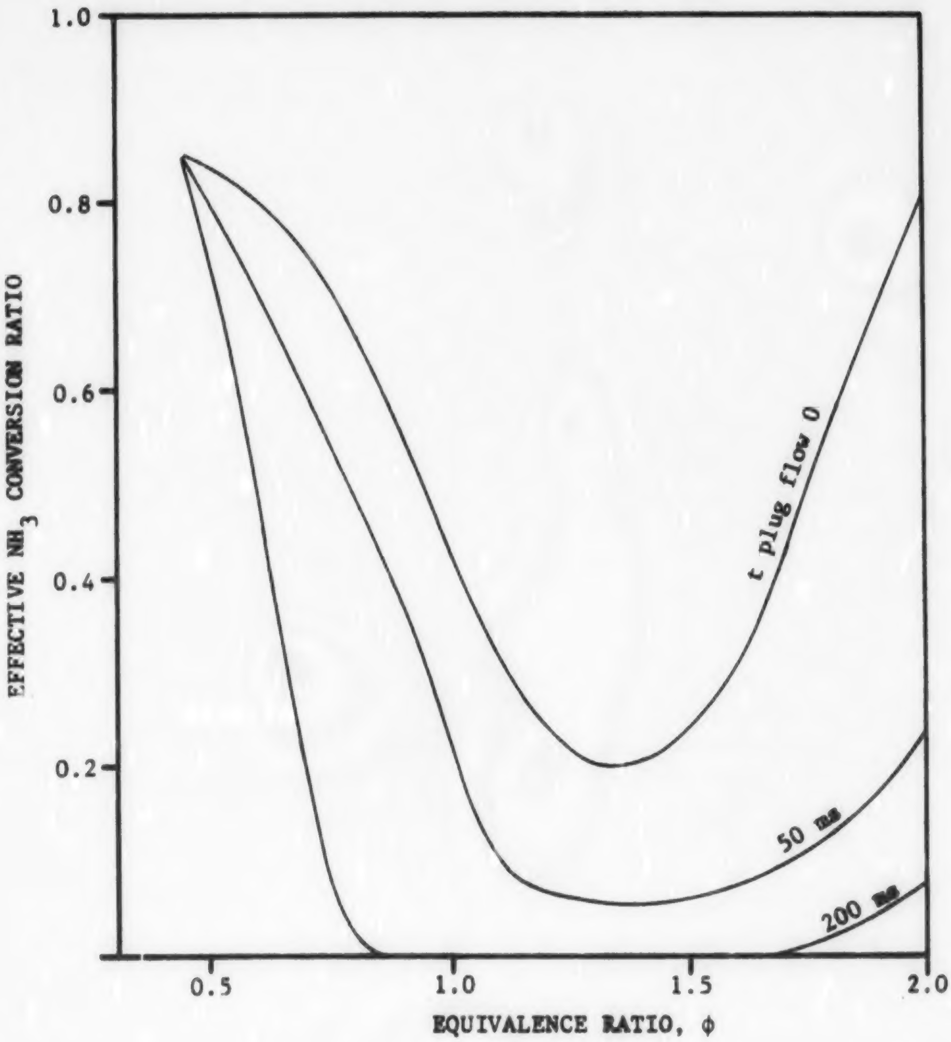

Figure 3. Effective NH_3 conversion ratio for premixed primary reactor - adiabatic gas turbine combustor.

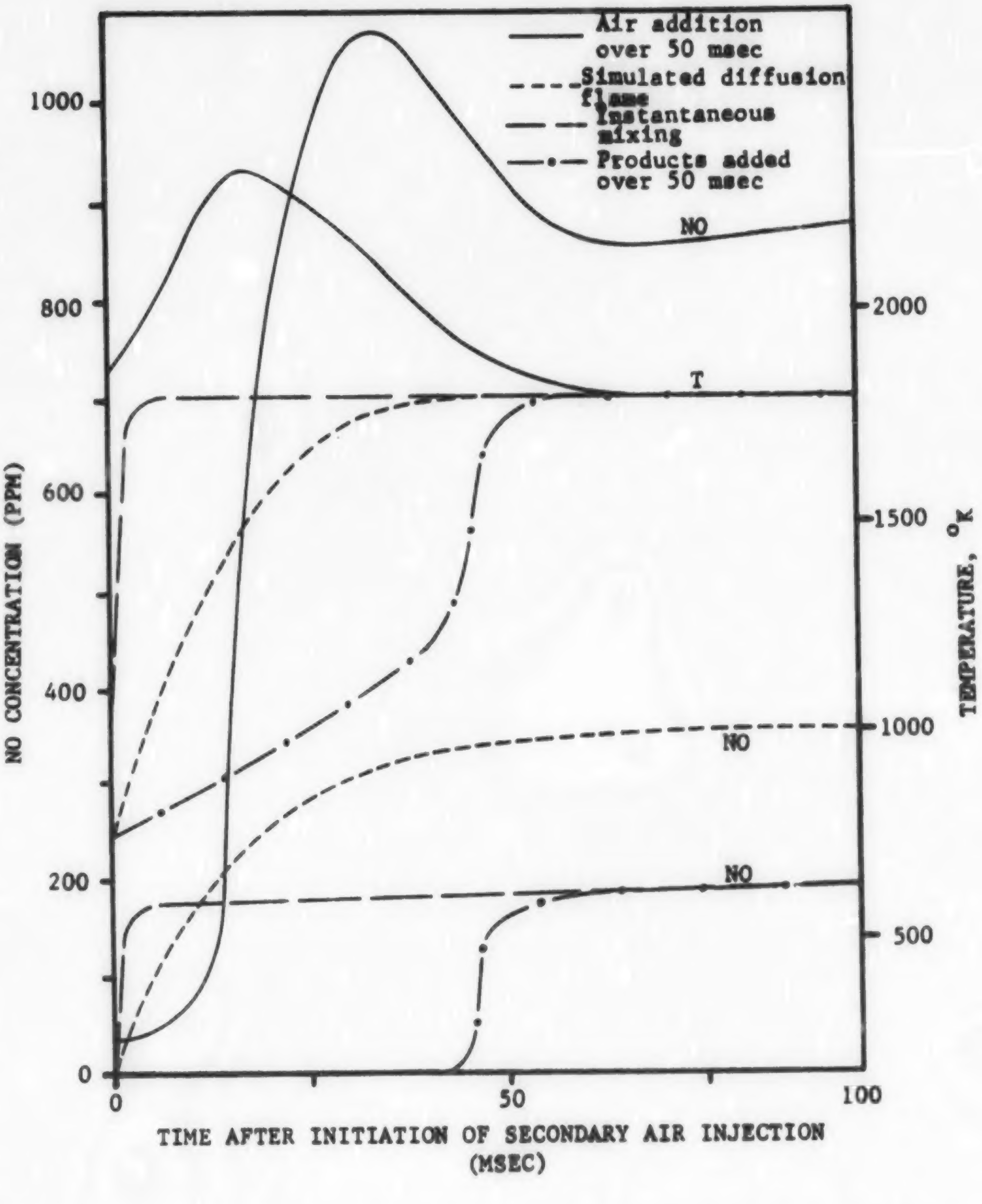

Figure 4. NO concentration and temperature for alternative secondary stage configurations - adiabatic gas turbine combustor (10 atmospheres).

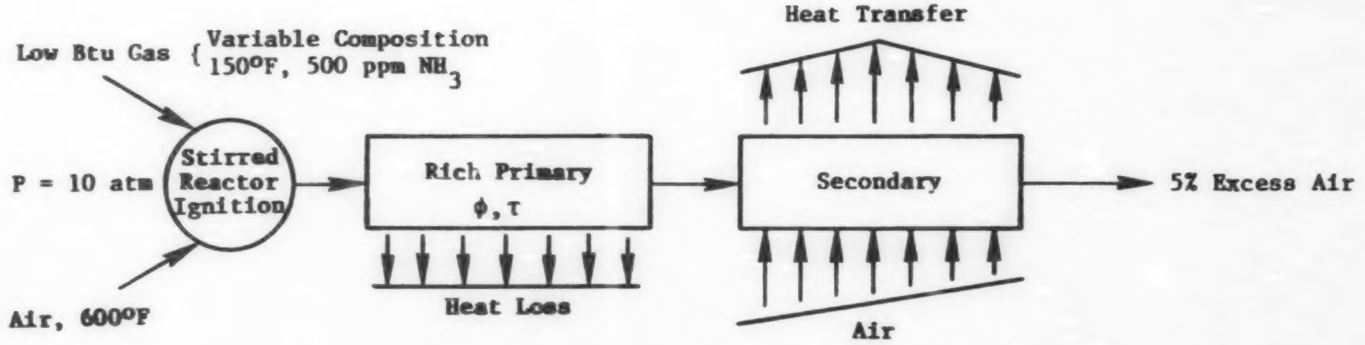
Low Btu Gas
Variable Composition
150°F, 500 ppm NH_3
P = 10 atm
Stirred Reactor Ignition
Air, 600°F
Rich Primary
ϕ, τ
Heat Loss
Heat Transfer
Secondary
Air
5% Excess Air

Figure 5. Limit-case analogue for the supercharged boiler.

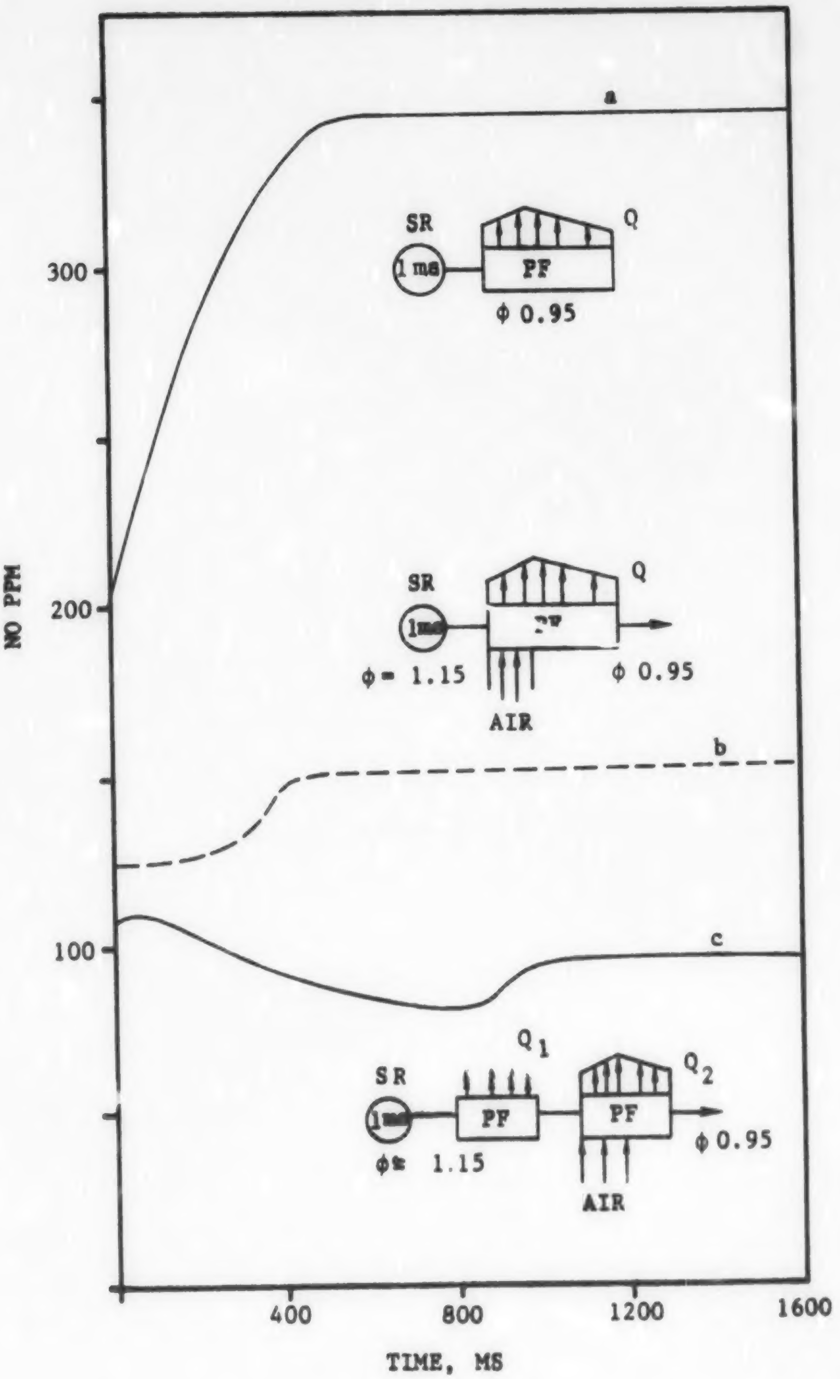

Figure 6. NO concentration for various reactor simulations of a supercharged boiler.

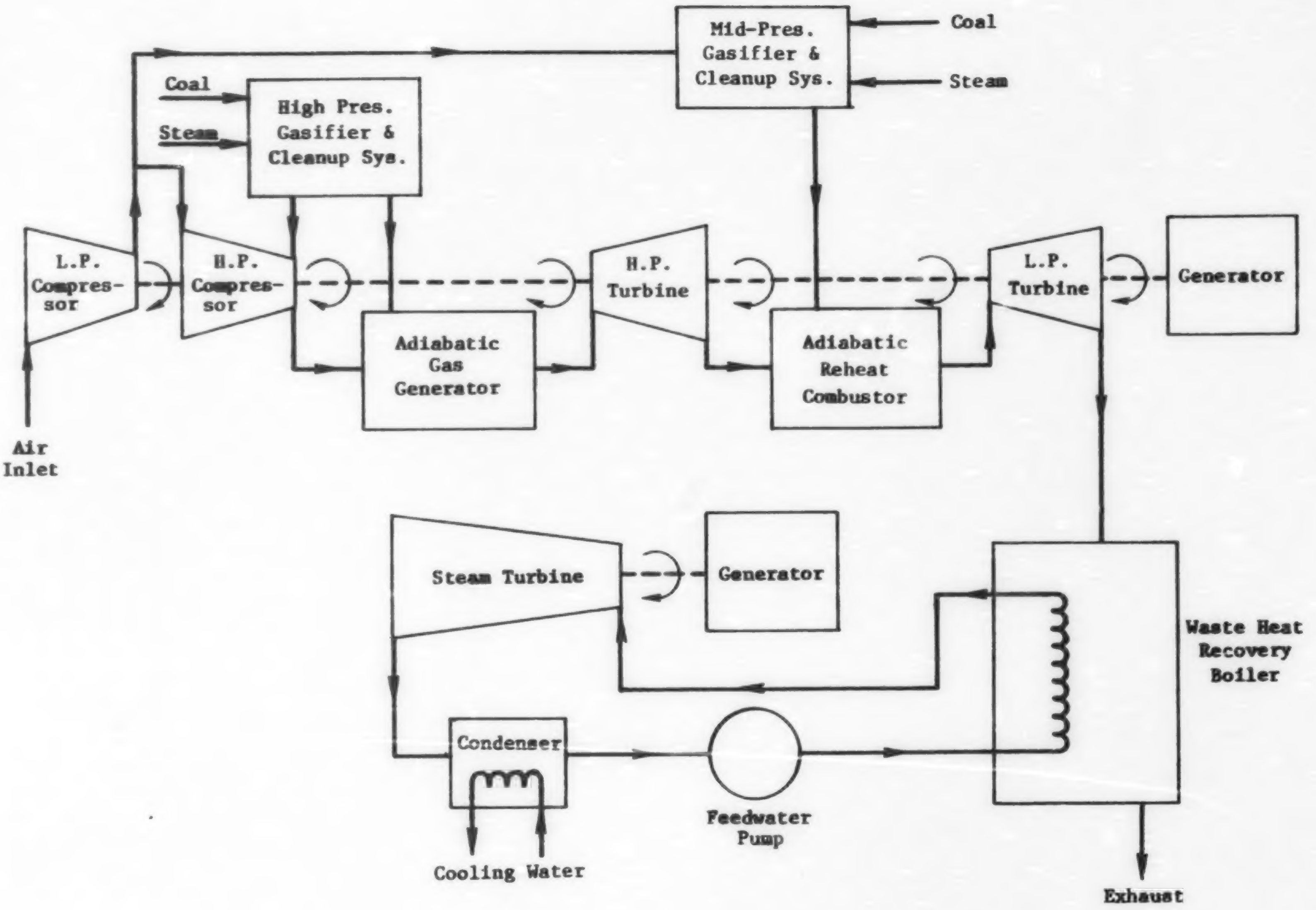
Mid-Pres.
Gasifier &
Cleanup Sys.
Coal
Steam
Coal
Steam
High Pres.
Gasifier &
Cleanup Sys.
L.P.
Compres-
sor
H.P.
Compres-
sor
H.P.
Turbine
L.P.
Turbine
Generator
Adiabatic
Gas
Generator
Adiabatic
Reheat
Combustor
Air
Inlet
Steam Turbine
Generator
Waste Heat
Recovery
Boiler
Condenser
Feedwater
Pump
Cooling Water
Exhaust

Figure 7. COGAS cycle with adiabatic main and reheat combustors.

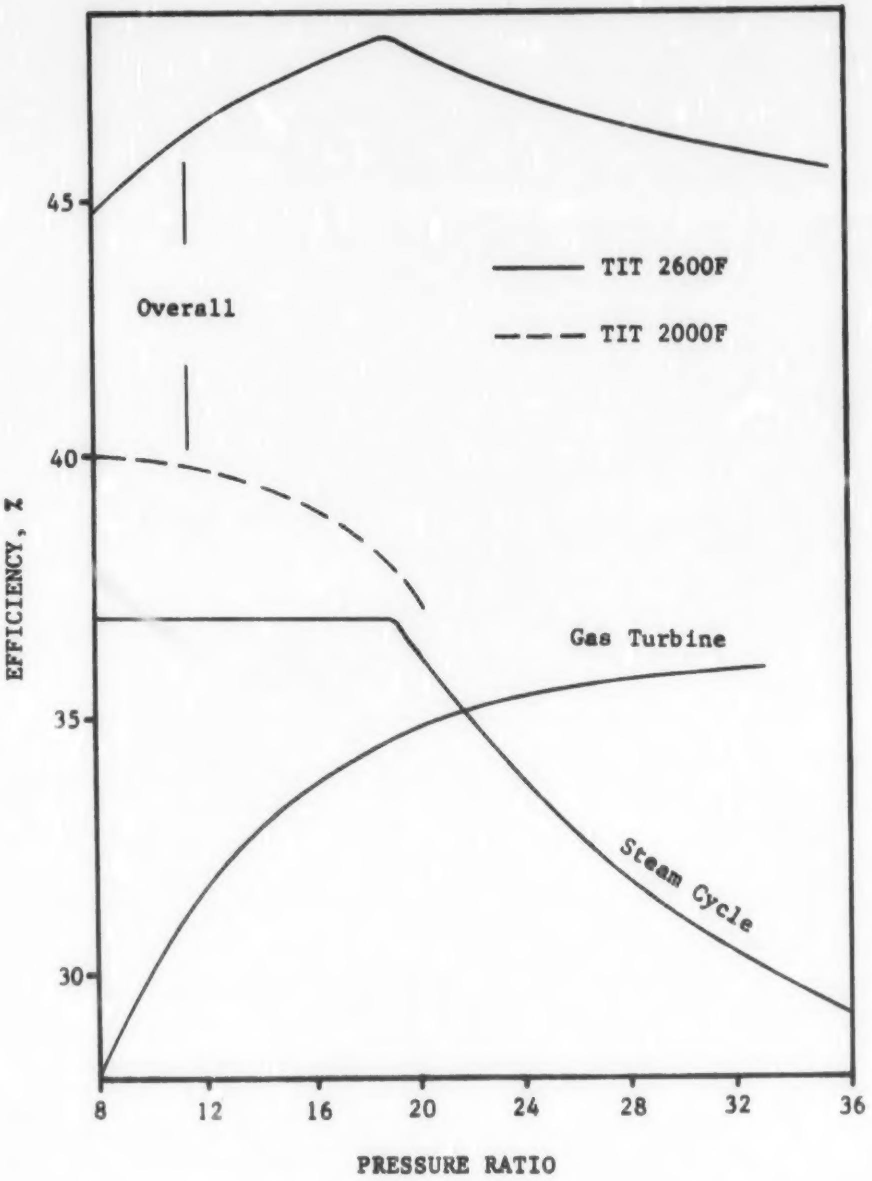

Figure 8. The influence of pressure ratio on cycle performance basic COGAS cycle.

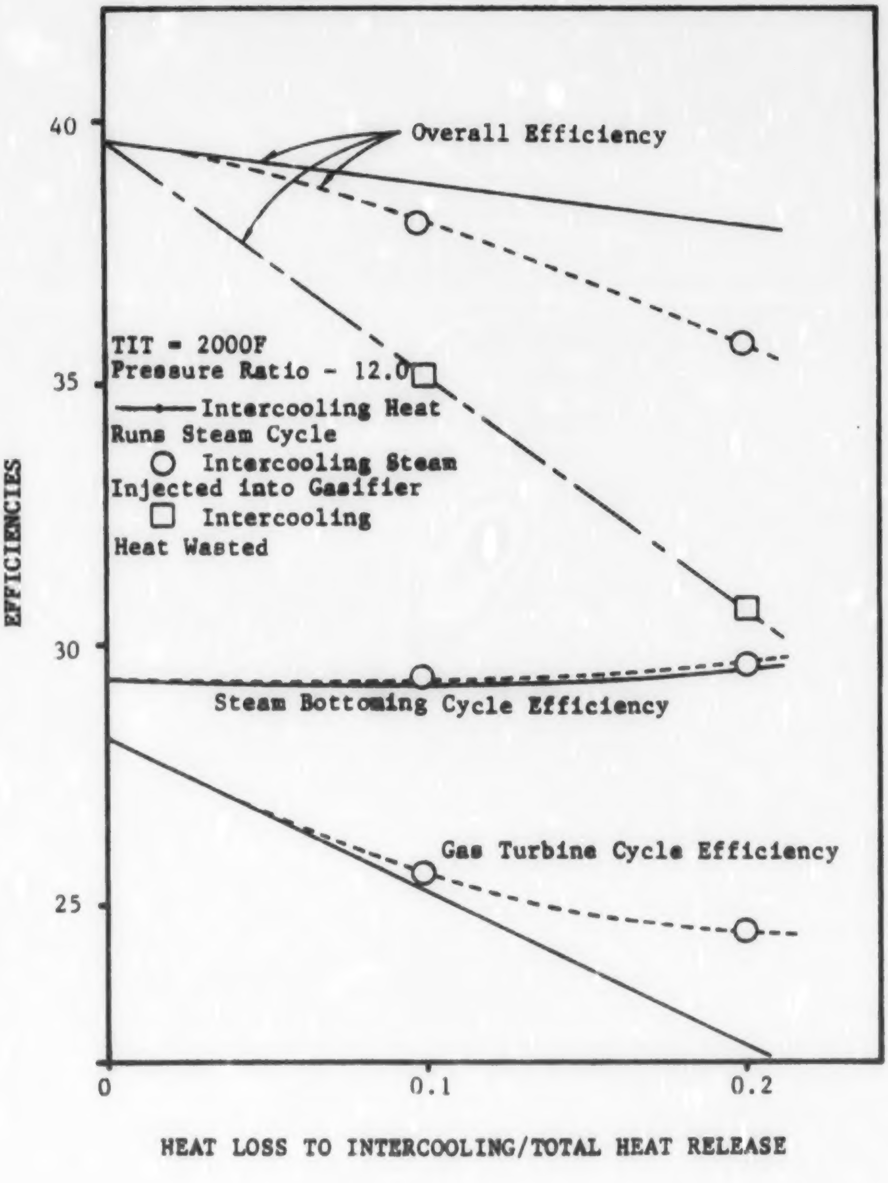

Figure 9. Basic COGAS cycle with gasifier intercooling TIT = 2000°F.

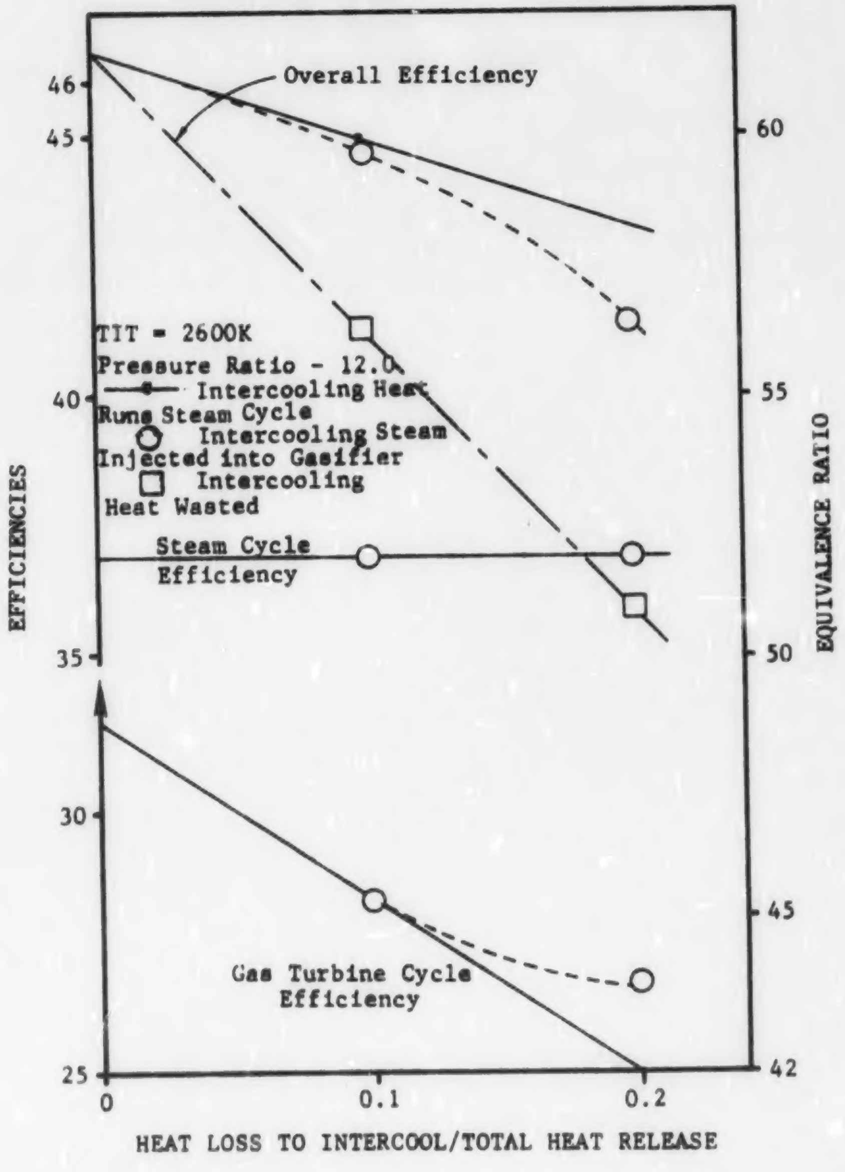

Figure 10. Basic COGAS cycle with gasifier intercooling TIT = 2600°F.

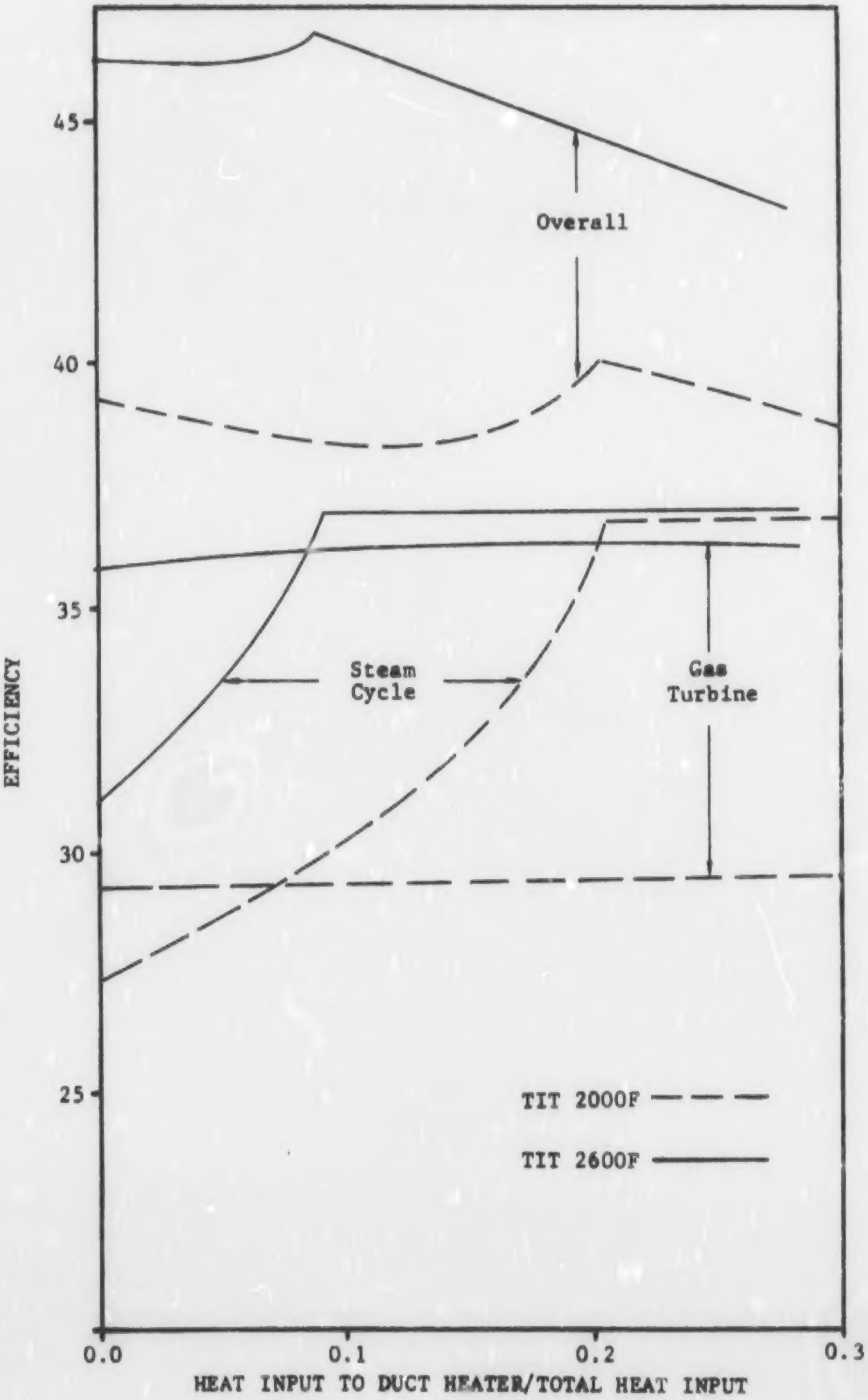

Figure 11. Basic COGAS cycle with duct heater (supplementary fired waste heat boiler).

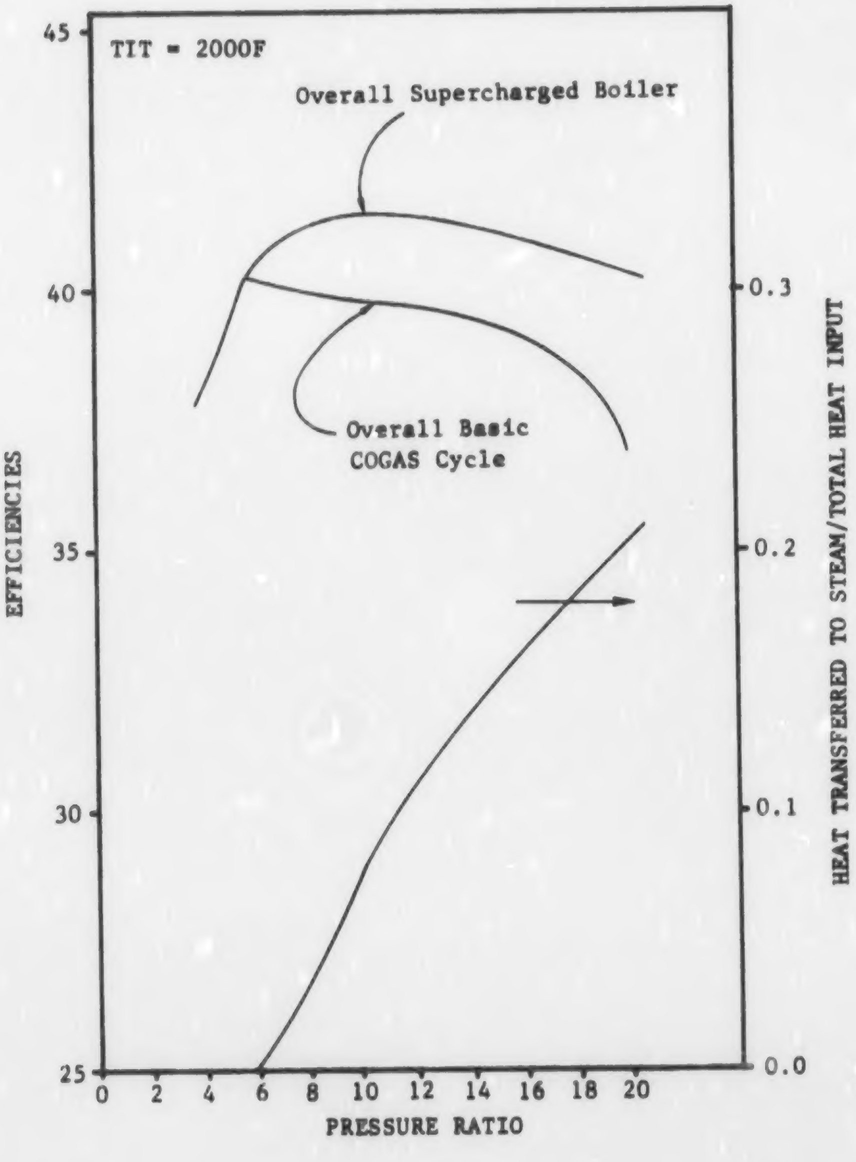

Figure 12. Comparison of basic COGAS cycle with the optimum supercharged boiler - TIT = 2000°F.

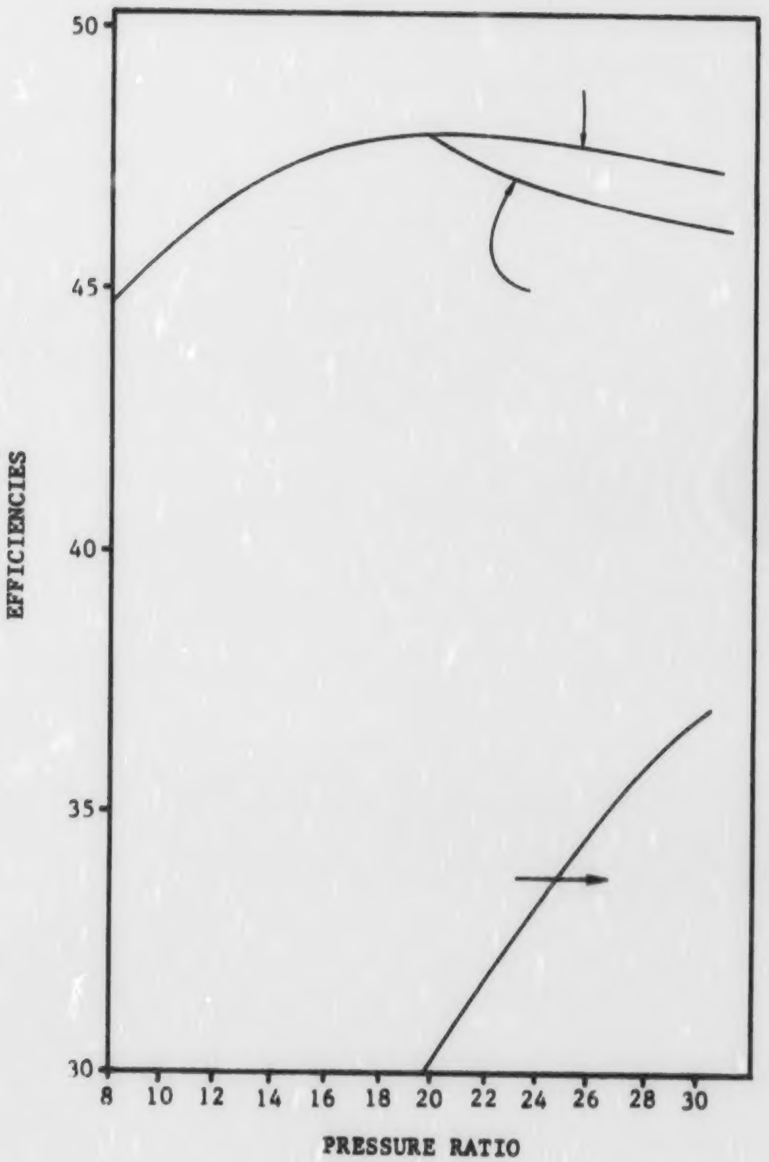

Figure 13. Comparison of basic COGAS cycle with the optimum supercharged boiler - TIT = 2600°F.

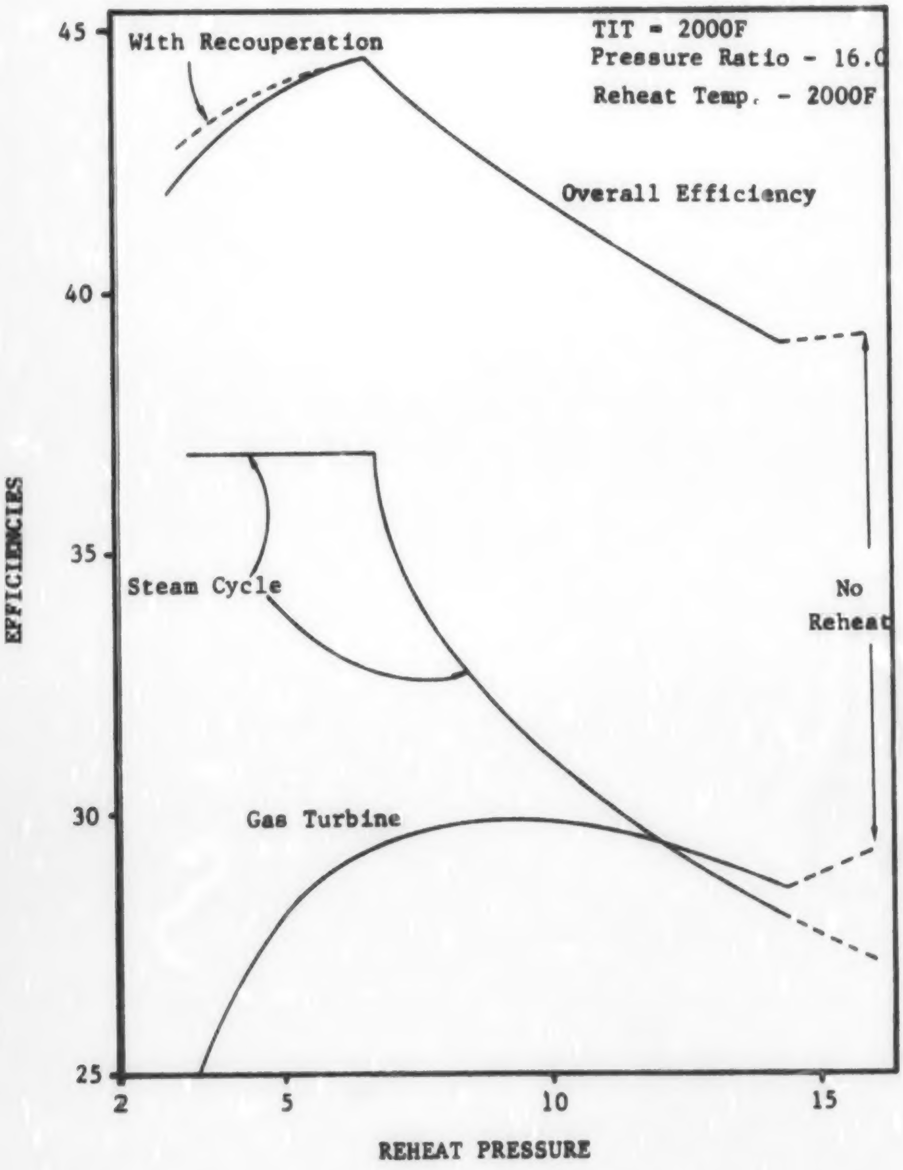

Figure 14. COGAS cycle with reheat-efficiency as function of reheat pressure.

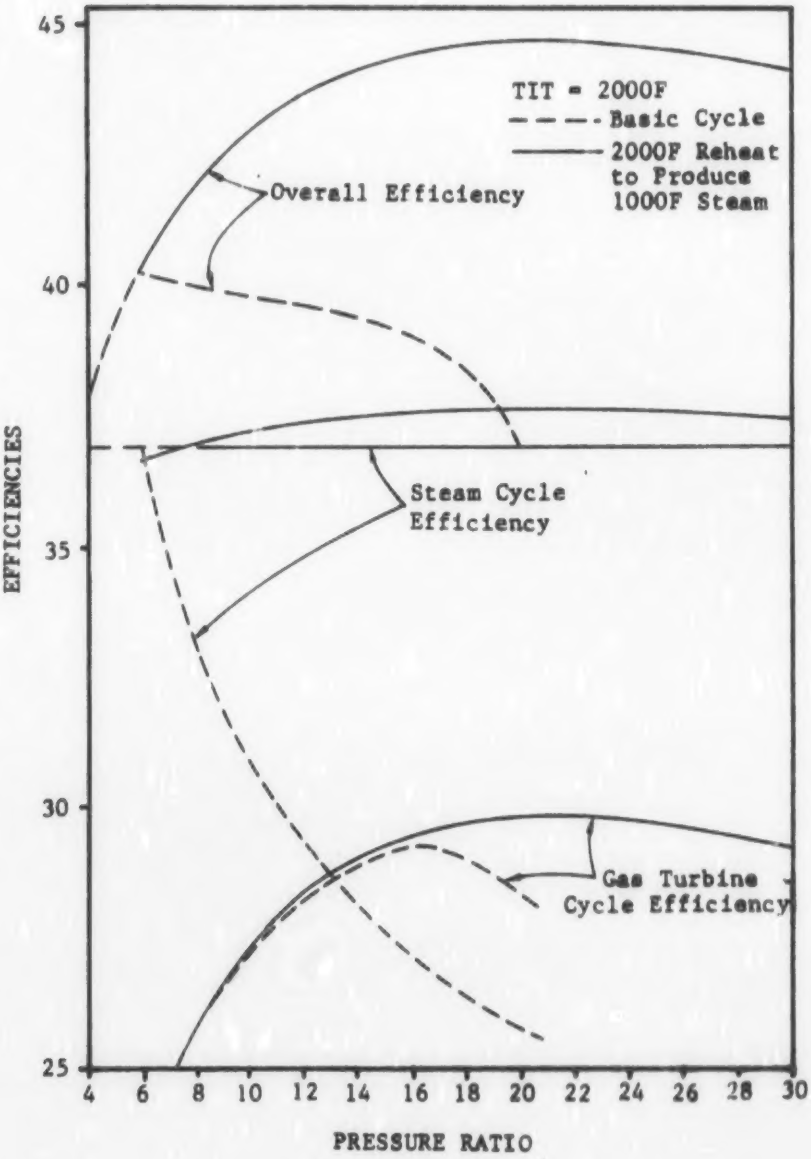

Figure 15. Comparison of basic COGAS cycle with reheat cycle as a function of pressure ratio.

PANEL:

EMERGING COMBUSTION TECHNOLOGIES

1 -- Fluidized Bed Combustion
John M. Connell

2 -- Coal-Oil Mixture Combustion Technology
Casters B. Foster

3 -- Advanced Combined Cycles
Fred L. Robson

FLUIDIZED BED COMBUSTION

By:

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FLUIDIZED BED STEAM GENERATOR

The fluidized combustion of coal in a limestone bed is a promising, near time method for burning high sulfur coal in a utility plant while meeting present environmental standards. A 150 MW capacity plant is presently in the initial design phases and should present plans be implemented, the construction of the first large demonstration unit will start in 1981.

Coal and limestone fed onto a grid plate is burned in a fluidized condition and the sulfur oxide produced combines with the calcium to form calcium sulfate, a solid, which is removed with the ash. Since the temperature of the burning of the products of combustion is held to 1550F by the inclusion of heating surface within the bed the maximum conversion of sulfur dioxide to calcium sulfate is accomplished with a minimum conversion of nitrogen to NO_x.

During the last few years this technology has emerged from the laboratory and there is now the largest fluidized bed steam generator in service that has been built under the auspices of the Environmental Protection Agency at Rivesville, West Virginia. This unit is designed to produce 300,000 lb/hr of steam at 925F and 1350 psi for a rated electrical capacity of 30 MW. Data obtained from this installation is being used for design criteria assurance on the 150 MW Demonstration Unit now being designed.

This 150 MW Demonstration Plant includes four (4) main fluidized beds, each including heating surface within the bed and the gases emitted therein pass over additional heating surfaces in the economizer and air heater. Particulate matter is captured in dust collectors and is to be reinjected in a carbon burn up cell to complete the combustion of carbon particles elutriated from the main cells. A baghouse located after the air preheater will capture virtually all of the particulate matter before the gasses are discharged to the atmosphere.

The unit was designed by Foster Wheeler Energy Corporation for 1,100,000 lb/hr of steam at the supercritical pressure of 3800 psi and 1000F at the superheater outlet. Additionally there is 960,000 lb/hr of steam reheated to a 1005F. The unit will burn 133,000 lb/hr of coal (10,490 BTU/lb, 4.5% sulfur) using 57,000 lb of limestone per hour to capture as a solid over 90% of the sulfur in the fuel. The 30 MW Demonstration plant at Rivesville and the new 150 MW Demonstration plant described above will provide industry with a high confidence level for building still larger atmospheric fluidized bed steam generators in the 600 MW - 800 MW capacity range during the next decade.

COAL-OIL MIXTURE COMBUSTION TECHNOLOGY

By:

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Department of Energy
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Washington, D.C. 20545

COAL-OIL MIXTURE COMBUSTION TECHNOLOGY

C. B. Foster
Manager, ERDA Coal-Oil Mixture Combustion Program
Energy Research and Development Administration
Washington, D.C. 20545

The Energy Research and Development Administration (ERDA) has undertaken in a cost-sharing partnership with industry and the utilities, an extensive coal-oil mixture combustion program in a number of promising applications. These applications currently include utility steam generators, industrial steam generators, and blast furnace injection, and can be expected to expand to other applications in the future. The objectives of ERDA's program are to modify and retrofit, operate and test existing boilers and blast furnaces, and to design, construct, operate and test prototype coal-oil mixture preparation facilities to demonstrate coal-oil mixture technology. The intent is to substitute coal for an appreciable fraction of oil in the near-term time frame and determine the economics of coal-oil mixture preparation, transportation and combustion. The coal-oil mixture combustion program is still in the early stages of development. However, some test results have been obtained and will be discussed as well as the economics of implementation and plans for future testing.

AVANCED COMBINED CYCLES

By:

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United Technologies Research Center
East Hartford, Connecticut 06108

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EMERGING COMBUSTION TECHNOLOGIES: ADVANCED COMBINED CYCLES

The increased use of coal in utility power systems has become a part of the national energy policy. To do this in an environmentally acceptable manner will require some type of pre- or post-combustion treatment. When advanced power systems such as gas turbine-based combined-cycle systems are considered, one potentially attractive method of coal treatment is the gasification of coal followed by desulfurization with the clean fuel gas being burned in the gas turbine.

A schematic of the combined gas-steam turbine system is given in Fig. 1. Fuel gas from the gasifier is burned in gas turbine and the combustion products expanded through the turbine which drives a generator. The hot exhaust products are then sent to a waste-heat recovery boiler which generates steam for a steam turbogenerator set. When operating at temperatures of 1425 C (2600 F) overall system efficiencies of 42-44 percent could be realized.

Such power systems would, of course, have to meet environmental standards. At this time, the only standards which have been promulgated are those for coal-fired steam stations (Fig. 2). However, standards for gas turbine-based power systems have been suggested and are included in Fig. 2 for comparison purposes.

Because the sulfur bearing compounds are removed in the fuel gas cleanup system prior to combustion, the major pollutant of interest is NO_x. In order to better understand the potential for NO_x pollution, the Power Systems Division of UTC and the Texaco Development Company carried out joint tests at the Texaco test facilities in Montebello, California. A small gasifier using residual oil, coal, and mixtures of the two was used to provide fuel to a small-scale gas turbine combustor (1/8 of a 30-MW gas turbine). Both air and oxygen were used in the gasifier so that fuel gases of approximately 900 kcal/M^3 (100 Btu/ft^3) and 2700 kcal/M^3 (300 Btu/ft^3) respectively, were produced.

The results of these tests using conventional combustor nozzles are shown in Fig. 3. Also shown, for comparison, are the NO_x emissions from natural gas (8900 kcal/M^3). It is apparent that at the burner temperature rise required for high-temperature turbine operations, i.e., 800 C and beyond, the emissions from the medium-heating value fuels could very well exceed proposed future regulations. The emissions from the low-heating value fuel appear acceptable, but further reduction would make the concept even more attractive.

One method of reducing NO_x involves shortening the time at which the fuel/air mixture is at high temperature. Since mixing time is the governing time ($\sim$10 msec for turbulent mixing vs 0.3 - 0.5 msec for combustion) in the conventional combustor, significant reduction in time would be achieved if the fuel and air were mixed prior to introduction into the combustor. This also would allow combustion at other than stoichiometric conditions usually present in conventional burners. These advantages are shown in Fig. 4 where emissions are shown as a function of equivalence ratio $\left(\frac{f/a\ \text{actual}}{f/a\ \text{stoichiometric}}\right)$ and residence time in the primary zone of the combustor. Here it is seen that combustion at a lean equivalence ratio (0.4 to 0.5) results in extremely low NO_x emissions, less than 25 ppm with low-heating value gas.

This was borne out in tests in which the medium-heating value gas was burned in a premix combustor (Fig. 5). Here the emissions with a premix combustor are from one-quarter to one-half that of the conventional combustor. When steam is injected as a NO_x suppressant, an even greater reduction is obtained.

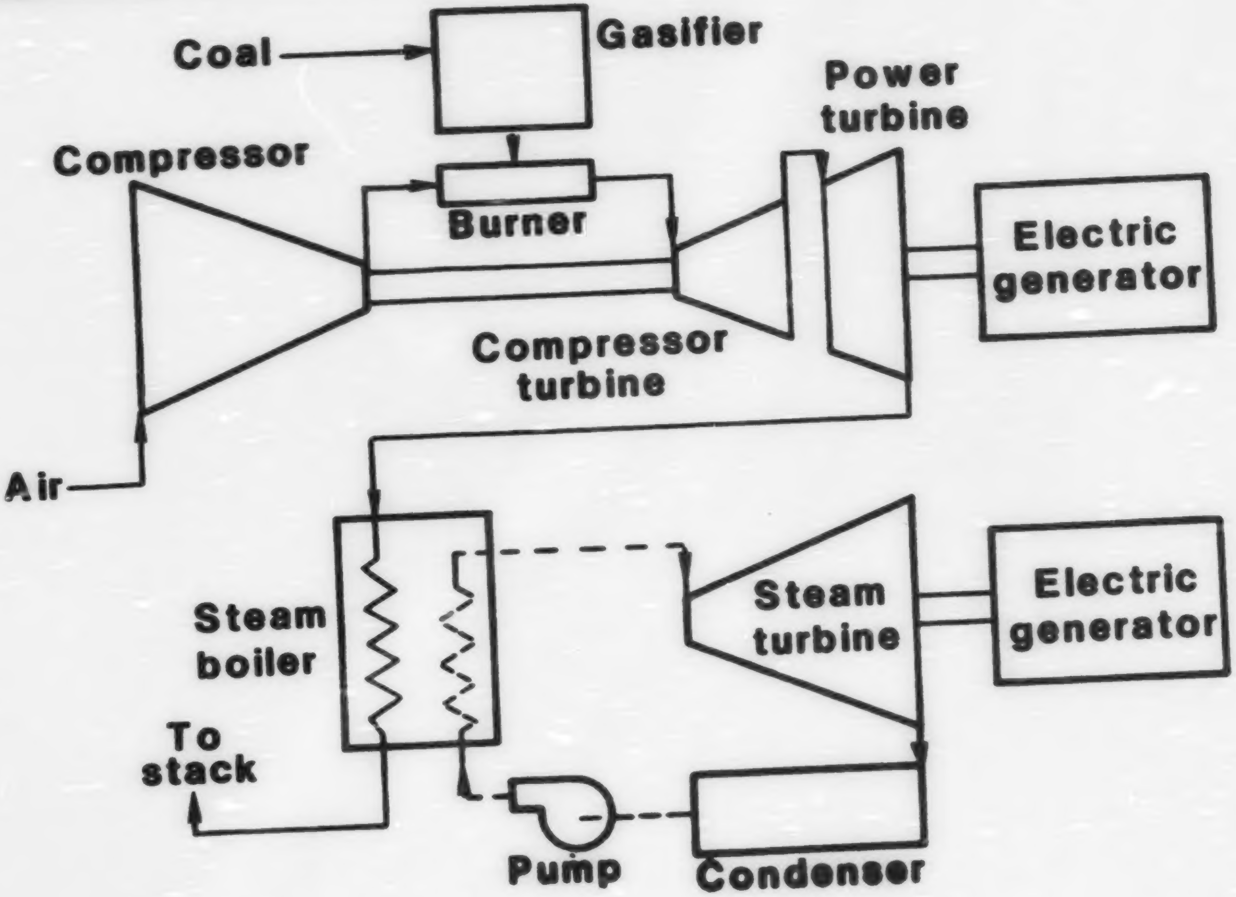
A potential configuration of a pre-mixed combustor is shown in Fig. 6. Because of combustor cooling considerations inherent when the gasifier is integrated with the gas turbine (15-17 percent of the air goes to the gasifier) a combustor having a low surface to volume ratio is necessary This results in the minimum amount of combustor wall to be cooled. An annular combustor, i.e., a single combustor wrapped around and concentric to the frame of the gas turbine, appears attractive. The combustor head is comprised of many small tubes through which the air is admitted to the combustor. Each tube has holes located radially around them through which fuel gas is admitted to the tube. Mixing takes place in the remainder of the tube prior to admission to the primary zone of the combustor.

Based upon the forgoing combustor concept, estimates of NO_x emissions have been made and are shown in Fig. 7. Here it is seen that the NO_x emissions are well below the current coal-fired power system rule, which is just met by well designed conventional steam systems.

Because SO_2 rules have been proposed for gas turbines, the projected emissions of SO_2 are given in Fig. 8. The emissions from the combined cycle vary as a function of the cleanup method, but are still significantly lower than the regulations, both current and projected, as well as lower than conventional steam with fuel gas desulfurization.

Figure 1

COMBINED GAS-STEAM TURBINE SYSTEM


Coal
Gasifier
Power turbine
Compressor
Burner
Electric generator
Compressor turbine
Air
Steam boiler
Steam turbine
Electric generator
To stack
Pump
Condenser

R11-163-1

Figure 2

Emission Standards for Coal-Fired Power Plants

	Conventional plant	Proposed gas turbines*
SO_2	0.57 kg/GJ (1.2 lb/10^6 Btu)	100 ppm
NO_x	0.33 kg/GJ (0.7 lb/10^6 Btu)	75 ppm
Particulates	0.047 kg/GJ (0.1 lb/10^6 Btu)	NA

* For all fuels and at ISO conditions with 15% O_2 in exhaust

77-08-159-3

NO_x Emissions-Combustors Tests on LHV and MHV Fuels

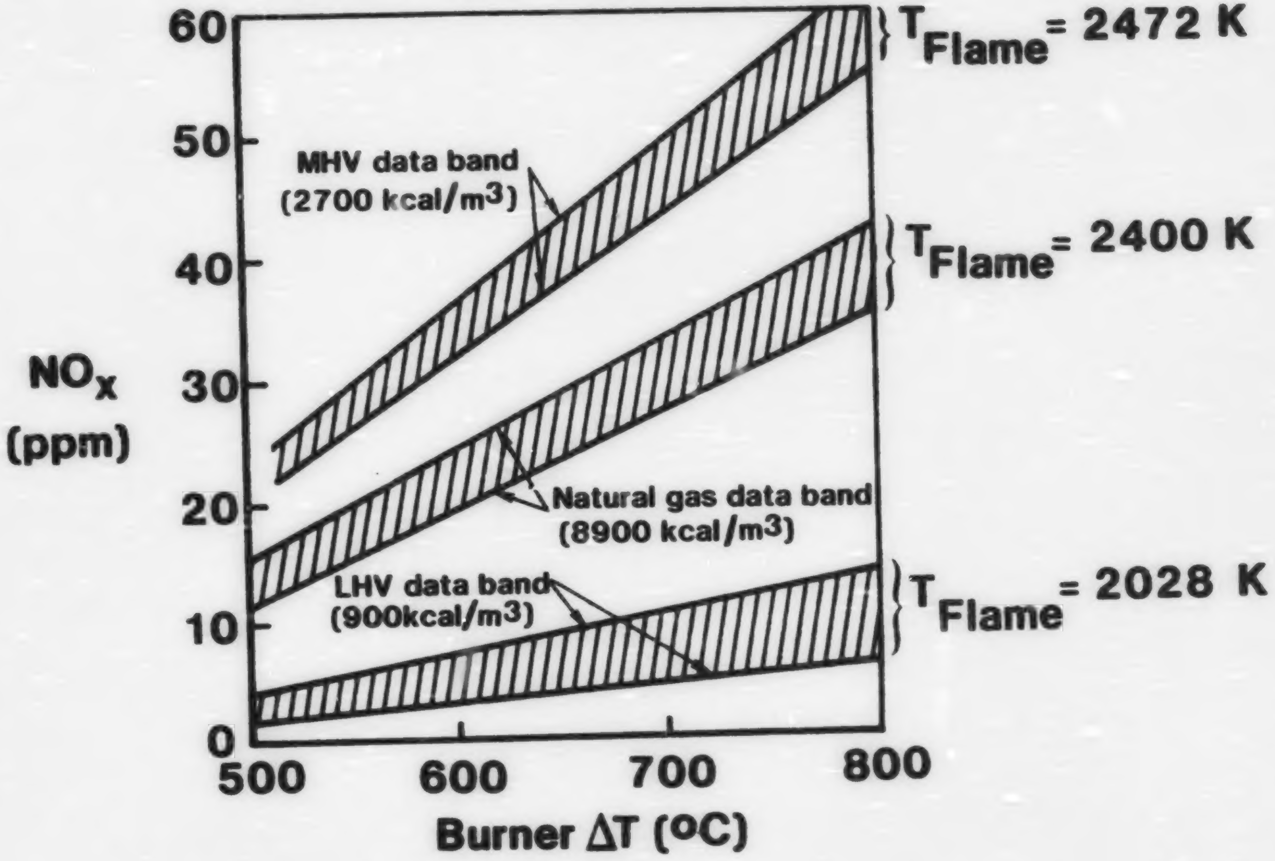


Figure 3

77-08-139-3

NO_x Estimation for Premixed Combustion

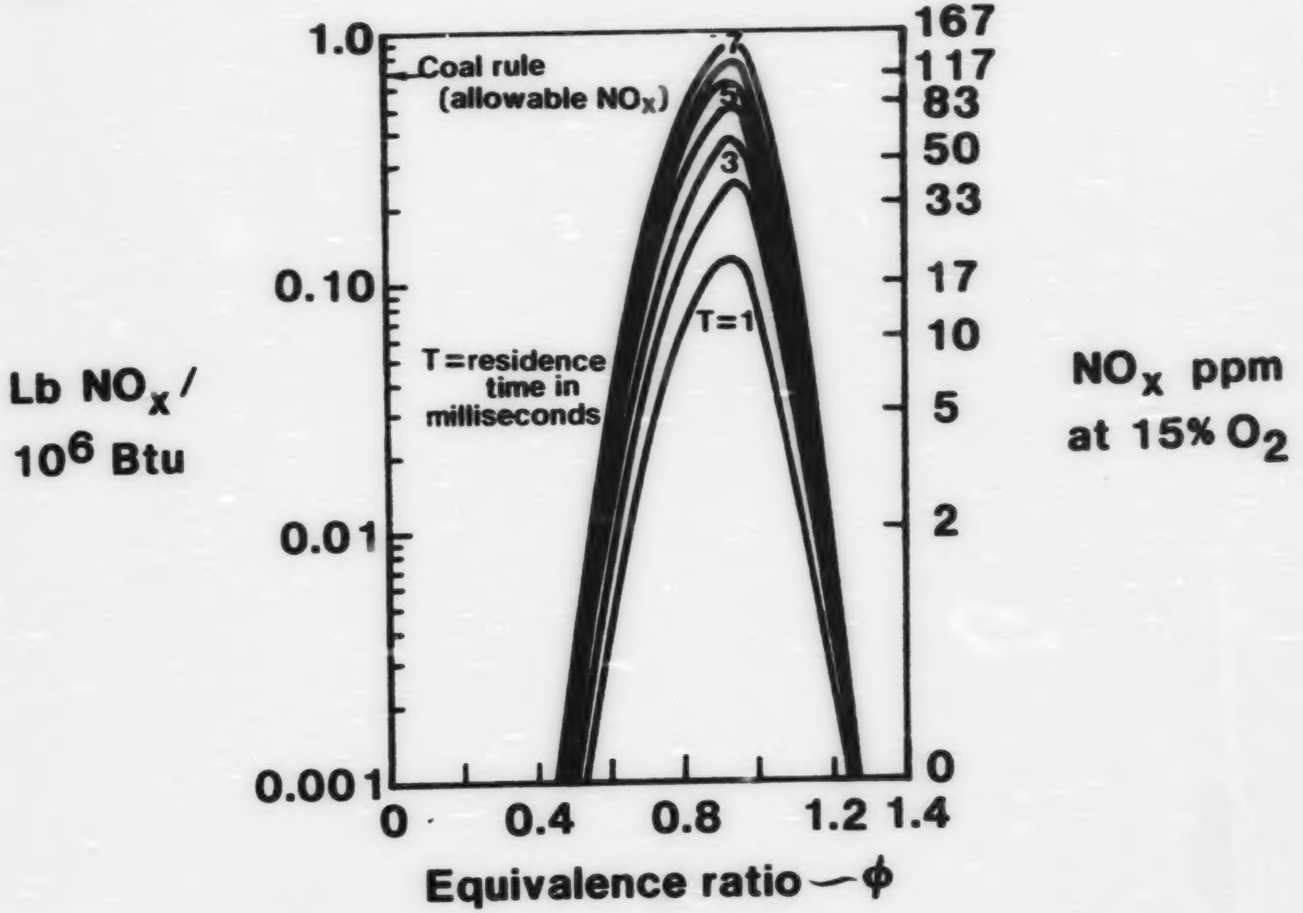


Figure 4

77-08-139-1

NO_x Emissions-Combustor Tests on MHV Fuel

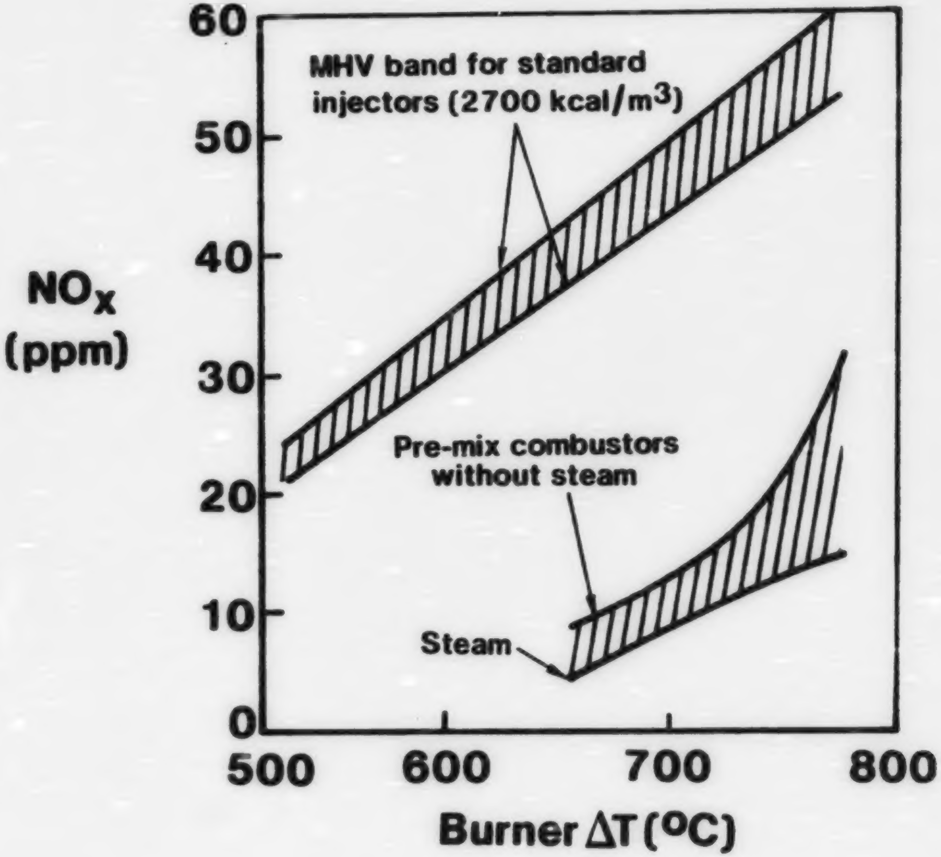
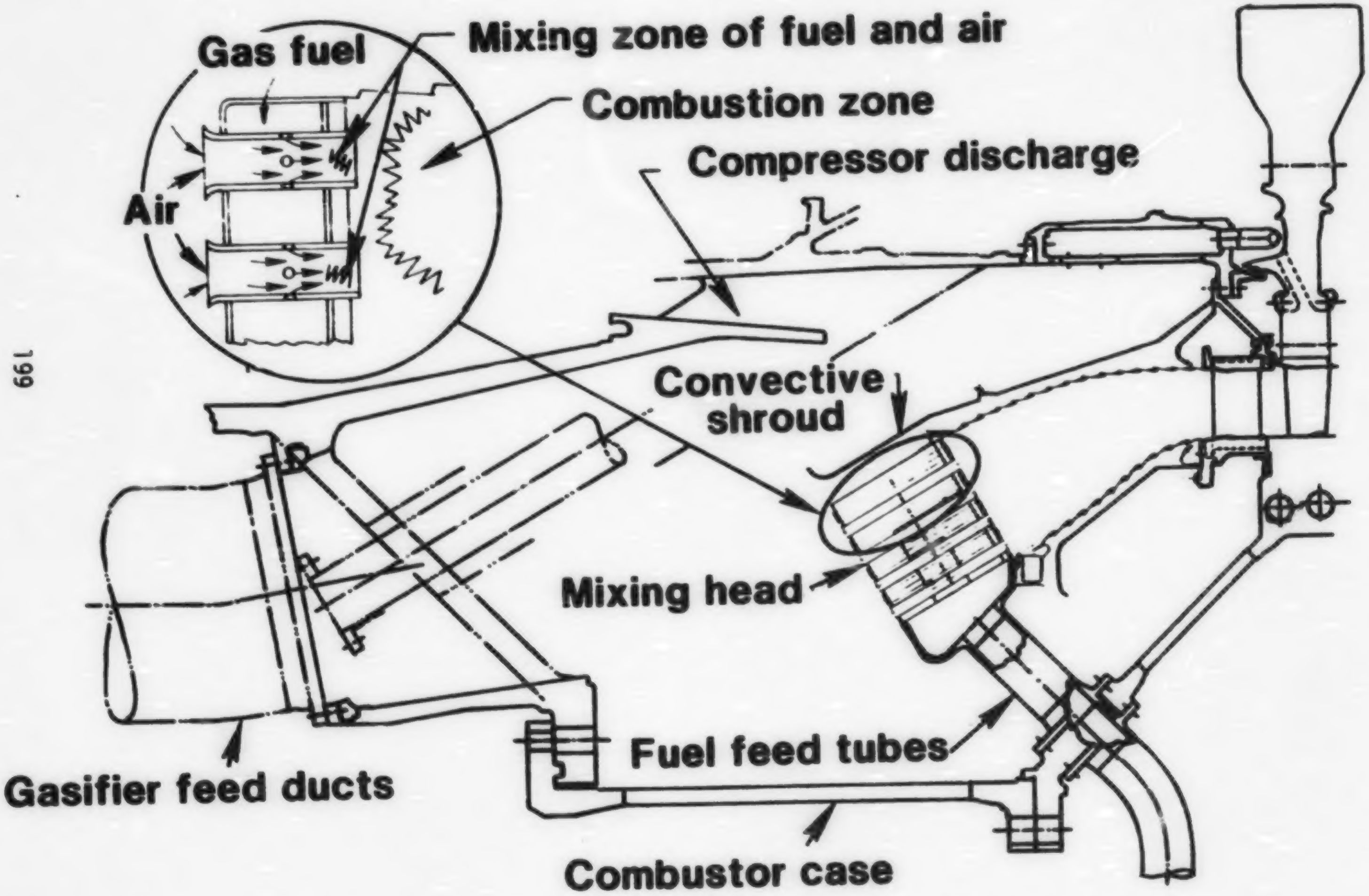


Figure 5

77-08-139-2

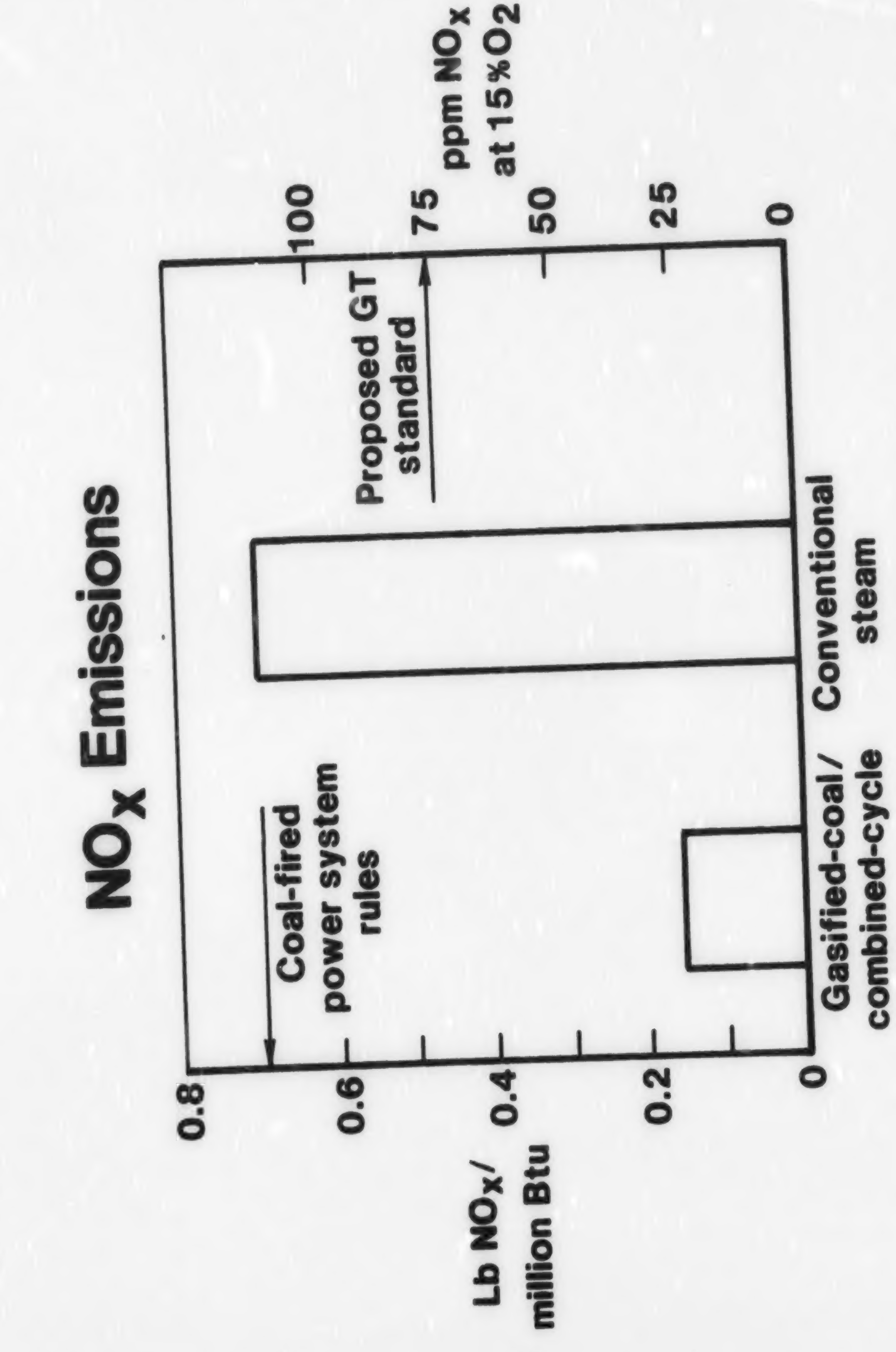
Figure 6

Potential Pre–Mix Combustor Layout


Gas fuel
Mixing zone of fuel and air
Combustion zone
Compressor discharge
Air
Convective shroud
Mixing head
Fuel feed tubes
Gasifier feed ducts
Combustor case

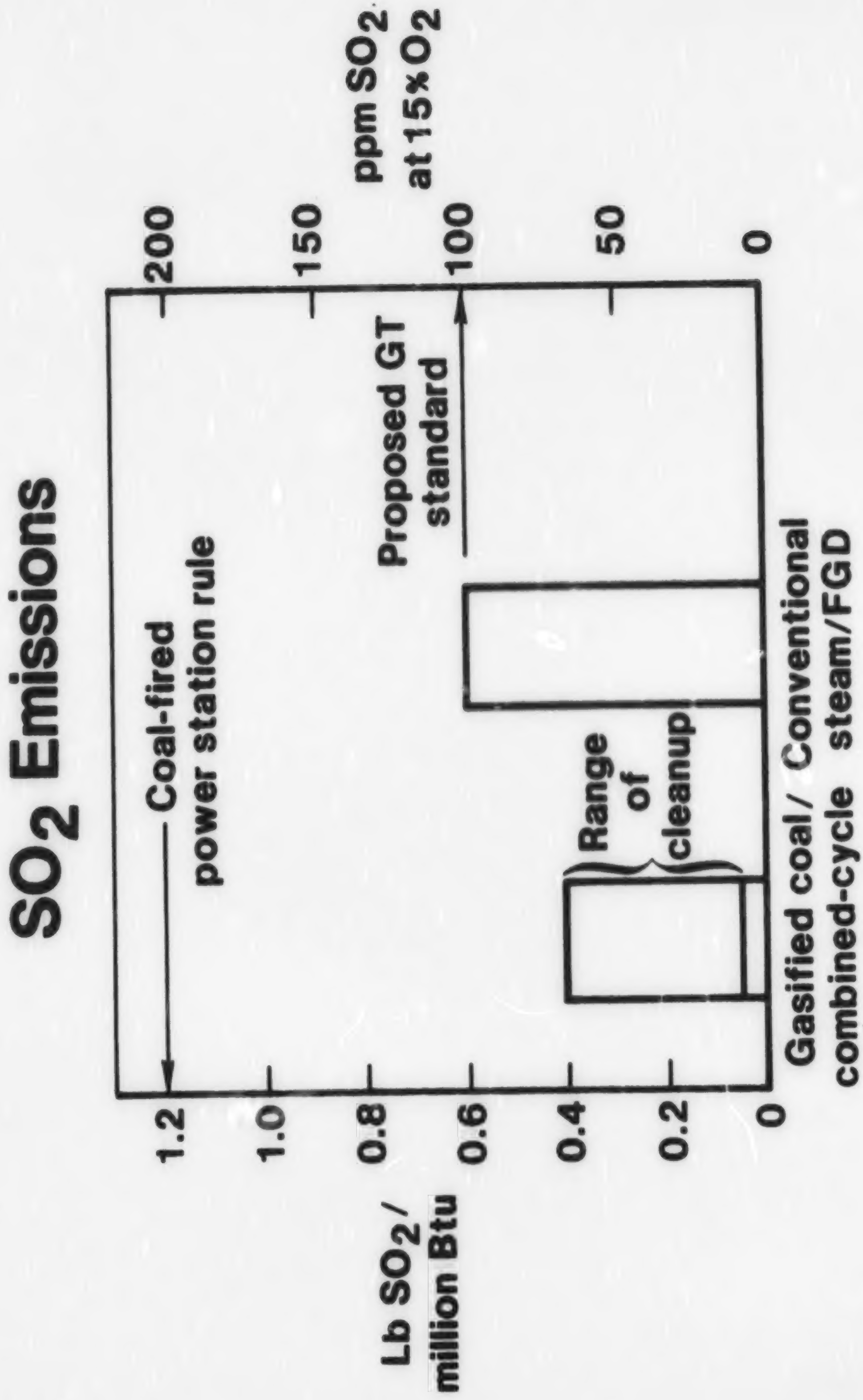
77–08–138–1

Figure 7



77-08-159-2

Figure 8



SO_2 Emissions

77-08-159-1

ADVANCED COMBUSTION SYSTEMS FOR STATIONARY GAS TURBINES

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This paper was not received in time for publication.

APPENDIX A:

LIST OF ATTENDEES

LIST OF ATTENDEES

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August 29 - September 1, 1977

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